# MACHINE DRAWING AND DESIGN

## A Textbook of Intermediate Standard for Engineering Students

BY

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O.B.E., Ph.D., B.Sc.(Hons.)Lond., M.I.Mech.E.

**Practical Geometry and Engineering Graphics.** A Textbook for Engineering Students

**An Introduction to Technical Drawing** A Complete Course for use in Secondary and Central Schools. In three Parts.

## By W. Abbott and W. Millar

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# PREFACE

This book is intended to provide an intermediate course in Machine Drawing and Design for students attending Technical Schools and Colleges. The work included is suitable for students taking Senior Courses in Engineering Drawing and Advanced Courses in Machine Design; also for students preparing for Professional and University Examinations in Mechanical Engineering. The inclusion of representative designs and of standard data will, it is hoped, also render the book of some service to draughtsmen and others engaged in engineering practice.

The book is arranged in two parts. In Part I the more important details of machinery are described and illustrated, and the principles underlying their design discussed. Part II is supplementary to the text of Part I, and consists entirely of exercises in drawing and design, graded in difficulty, together with representative questions from examination papers. These exercises are intended mainly for work at home, following a class-room lecture.

Most of the drawings are pictorial, and take the place of models used in earlier work. As far as practicable, they are so arranged that, although the shape and function of the object are at once apparent, some expenditure of thought is needed to produce the drawings required in the exercises: the mere copying of orthographic views has been avoided wherever possible. Minor dimensions and details have been omitted deliberately, particularly in the more advanced plates, to encourage initiative. More difficult orthographic drawings have been included to give practice in interpretation.

In preparing the text, it has been assumed that the allied subjects of Mechanics and Strength of Materials are being studied concurrently, so that the various formulæ used will not be unfamiliar: the inclusion here of definitions and proofs from Mechanics would extend the book considerably without greatly increasing its utility. Reference is made in the text to more advanced authoritative work to which the student may turn for further information.

The designs included have been selected not necessarily because they are the best, but because they are representative of modern and successful practice. The original designs from which the plates were prepared were invariably too complicated for the use of students, and although they have been simplified and amended great care has been taken not to sacrifice any important feature. No attempt has been made to include all varieties of design for the same part; neither have drawings of assembled engines or machine tools been given: these are best studied in works on general engineering. To those Firms, enumerated here, who have so generously supplied material and data, the author is deeply

indebted; whatever merit the book may possess is due in large measure to their assistance and co-operation.

Acknowledgment is made of the courtesy of the Senate of London University, of the Court of Glasgow University, and of the Controller, H.M. Stationery Office, in permitting the reproduction of questions from Examination Papers. Thanks are due to the British Standards Institution for allowing the inclusion of extracts from their reports, and to the Council of the Institution of Mechanical Engineers for permitting the use of material from their Proceedings.

The task of reading the proofs of the book was kindly undertaken by Mr. E. H. H. Gibbins, B.Sc.: to him, and to many other friends who have supplied useful information, the author's best thanks are tendered.

W. A.

1930.

Alphabetical List of Firms who have supplied designs and data:—

| | | |
|---|---|---|
| Messrs. | Sir Wm. Arrol & Co., Ltd. | Crane Parts. |
| „ | Sir W. G. Armstrong, Whitworth & Co., Ltd. | Locomotive Parts. |
| „ | Andrew Barclay, Sons, & Co. | Locomotive Parts. |
| „ | Beyer, Peacock, & Co., Ltd. | Locomotive Parts. |
| „ | David Brown & Sons (Hudd.), Ltd. | Gearing. |
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| „ | The Unbreakable Pulley and Millgearing Co., Ltd. | Bearings and Shaft Fittings. |
| „ | Worthington-Simpson, Ltd. | Steam-engine Parts. |

NOTE.—The use of Greek letters in the technical press, and in literature dealing with standardised data for gearing, has led the author to include the most common. They are: $\alpha$ (alpha), $\beta$ (bēta), $\theta$ (thēta), $\lambda$ (lambda), $\pi$ (pi), $\rho$ (rho), $\Sigma$ $\sigma$ (sigma), $\phi$ (phi), $\psi$ (psi), $\omega$ (omega).

## NOTE TO FIFTH EDITION

The book has been brought into line with the recommendations of B.S. 308, 1943, and additional matter on cams and on tolerancing has been included.

W. A.

1947

# CONTENTS

# INTRODUCTION

**MACHINE DRAWING** deals with the application of practical geometry to the representation of machines. By the use of conventions which have been found convenient in practice, the ordinary geometrical projections are amended, and amplified with notes and symbols, to indicate much more than the mere shape of the machine parts. Using this graphic language, and with the principles of projection as a basis, the designer of machinery expresses his ideas in a form intelligible to craftsmen. Two of the many types of drawings are: those which show the arrangement, or indicate the function, of a machine; and those which are to be used in the works for the actual manufacture of the parts. It is the latter that are chiefly discussed herein: they are known as working drawings.

With a view to securing uniformity in Drawing Office practice in Britain, the British Standards Institution has issued a report (B.S. 308) recommending the adoption of a number of conventions which were found to be generally acceptable in engineering works and educational institutions throughout the country. These conventions are included herein together with others that have become common in practice, either in this country or in the United States of America.

**Drawing Equipment.**—The components should be of the best quality procurable. It is better to build up a set of good instruments by buying a few pieces at a time rather than to purchase an inferior complete set. For the work in this book the following equipment is required: it should be regarded as the minimum.

Drawing Board, 23″ × 16″, either battened or three ply. Tee Square, blade 24″ long (*not* recessed into the stock). Set squares, transparent celluloid, 60°–10″, 45°–6″. Scales, boxwood, of convex section (see below). Compasses, 6″, with shouldered needle points, pen and pencil points, and lengthening bar. Dividers, 5″, with fine adjustment. Bow Pencil Compasses, 3″, with needle point. Pencil Spring Bows. Protractor, celluloid, semicircular. French curves, one or two of the oval type. Pencils, 3H, 2H for drawing, H, HB for lettering. Drawing Pins, Rubber, Paper, &c.

If drawings are to be inked in or traced the following are required in addition: Spring Bow Pen; Ruling Pen, 4½″.

Proportional Compasses will be found useful if much isometric projection is done.

For machine drawing the scales should have divisions graduated in eighths and sixteenths. Frequently the only scales available have divisions graduated in twelfths: these are satisfactory for scales such as ¾″ = 1 foot, but not for those of ¾ full size.

**Tracing.**—In a works drawing office the pencil drawings are copied on tracing cloth in Indian ink so that prints may be made for issue to the shops. The prints are taken off on sensitized paper much in the same way as photographic prints are taken from negatives. The technique of tracing cannot be explained fully here: only a few hints can be given.

Pin the tracing cloth glossy face uppermost over the drawing and stretch it by rubbing with a chalked cloth. The chalk neutralizes the greasy surface of the tracing cloth. Re-pin the tracing cloth and begin tracing at one upper corner. Complete the work in stages (the cloth usually stretches further and it is necessary to work over small areas). Make any erasions with special rubber. Apply colours on the reverse side.

**MACHINE DESIGN** consists in the application of scientific principles to the practical constructive art of engineering, with the object of expressing original ideas in the form of drawings. The designer needs a knowledge of the subjects of Mechanics, Mechanism, Strength of Materials and Metallurgy; he must also be familiar with workshop processes—casting, forging, machining—and workshop organization; for production in quantity he must be expert in allocating tolerances to dimensions and in the use of jigs and fixtures.

The evolution of a design may proceed along the following lines:—

(1) The original or basic idea. (A machine is visualized which will perform a certain operation.)

(2) The arrangement of the machine in outline.

(3) Preliminary analysis of the forces which will act in the several parts of the machine.

(4) Choice of materials and determination of the proportions of the principal parts.

(5) Preliminary design of the parts of the machine: (*a*) choice of materials, (*b*) determination of processes involved, (*c*) modifications based upon practical experience, and upon the need for production planning.

(6) Preparation of working drawings.

The principles of Mechanism and Mechanics are applied respectively in (2 and (3), while (4) and (5) involve a knowledge of Materials, Metallurgy, and Workshop Techniques and Organization.

In dealing with (4), (5), and (6) the designer has to consider many minor factors.

For example, we may suppose that the part is to be a casting.

The casting must be of suitable form, and projecting parts should be slightly tapered so that the pattern may be withdrawn easily from the mould. Changes of thickness should be gradual and sharp edges should be avoided. If possible, large masses and thin parts should not join together. Again, provision for lifting the casting, or supporting it temporarily, may be necessary.

Further, the designer has to select suitable materials from a large range of alloys, a few of which are given on p. 220: no less than twenty different materials are used in a modern turbine.

The proportions arrived at may require serious modification if the cost of the machine is likely to be prohibitive. The design may be cheapened by substituting castings in iron for forgings and machined parts, or by the use of fabrication by welding; by limiting the amounts of expensive metals used; by adopting standard parts; by eliminating surplus material in every way; and by making the design suitable for economic production.

Often the designer is in the unhappy position of being unable to state definitely the loads that the parts of his machine will have to bear; of dealing with stresses too complex for analysis; of using materials which may not have the properties assigned to them; and of having to make allowances for such uncertain effects as those due to corrosion, mishandling, and wear. It is evident, therefore, that experience, based upon a knowledge of successful practice, will usually play the chief part in the shaping of a design.

The student should analyse various designs and assess the many factors that have made them successful. In all calculations he should put accuracy first, ensuring that his arithmetical work is correct *and that his units are in agreement*: preventable errors cannot be tolerated in machine design.

# PART I

## MACHINE DRAWING

### PRINCIPLES OF PROJECTION

**Orthographic Projection.** — If lines are drawn to meet a plane from selected points on the contour of an object, the outline given on the plane is called the **projection** of the object on the plane, and the lines are known as **projectors.** If the projectors are perpendicular to the plane, an **orthographic** projection is the result, as in fig. 2; if the projectors converge to a point they give a **perspective** projection, while if they are parallel but inclined to the plane they give an **oblique** projection. In machine drawing, orthographic projection is almost exclusively used, the views being easily drawn and scaled. They have the disadvantage, in comparison with pictorial views, that they are not always easily interpreted, especially to one untrained in the system. Compare figs. 6 and 7.

Two **principal planes** are used in orthographic projection, one horizontal and one vertical, intersecting and dividing space up into four angles or quadrants, numbered as in fig. 3. The orthographic projections of an object situated in one of these angles on the vertical and horizontal planes give respectively its **elevation** and **plan.** To show these projections on a plane surface, the planes are opened out, or **rabatted,** about the line of intersection (called the $xy$ or ground line) until they coincide. For practical purposes, only the 1st and 3rd angles are used for projections; in the 2nd and 4th the views overlap after rabatment and cause confusion; 3rd angle projection is discussed on page 10.

**1st Angle Projection.**—To secure uniformity in Drawing Office practice, the British Standards Institution has recommended that 1st Angle Projection be adopted as the standard.

Fig. 1 shows a bracket in the 1st angle, projected on to the two principal planes and on to an additional plane mutually perpendicular to them. Three views of the object are thus obtained: plan, elevation, and end view (or side elevation). When the planes are opened out, the views take the positions shown in fig. 4 (drawn to a smaller scale). It will be noted that the auxiliary vertical plane may be turned about either $x_1y_1$ or $x_2y_2$ (fig. 1), so that the end view may have one of two positions on rabatment, as in fig. 4. The relative positions of the views in fig. 4 correspond to the projections in fig. 1, but in practice the views would be brought more closely together.

The line joining the plan and elevation of a point, also called a projector, must be perpendicular to $xy$; in fig. 4, $pp_1$ is perpendicular to $xy$; also $p_1p_2$ is perpendicular to $x_1y_1$.

It is a distinctive feature of 1st Angle Projection that each view appears on the side of the object remote from the face that it portrays: i.e. a top view or plan is placed beneath the elevation; an end view looking from the left is placed on the right; and so on.

### EXERCISES

(1) Draw, full size, an elevation, plan, and end view of the objects shown in figs. 5 and 6.

(2) Draw, full size, the views given in fig. 7 and project an additional end view. Then draw, freehand, a pictorial view of the object represented.

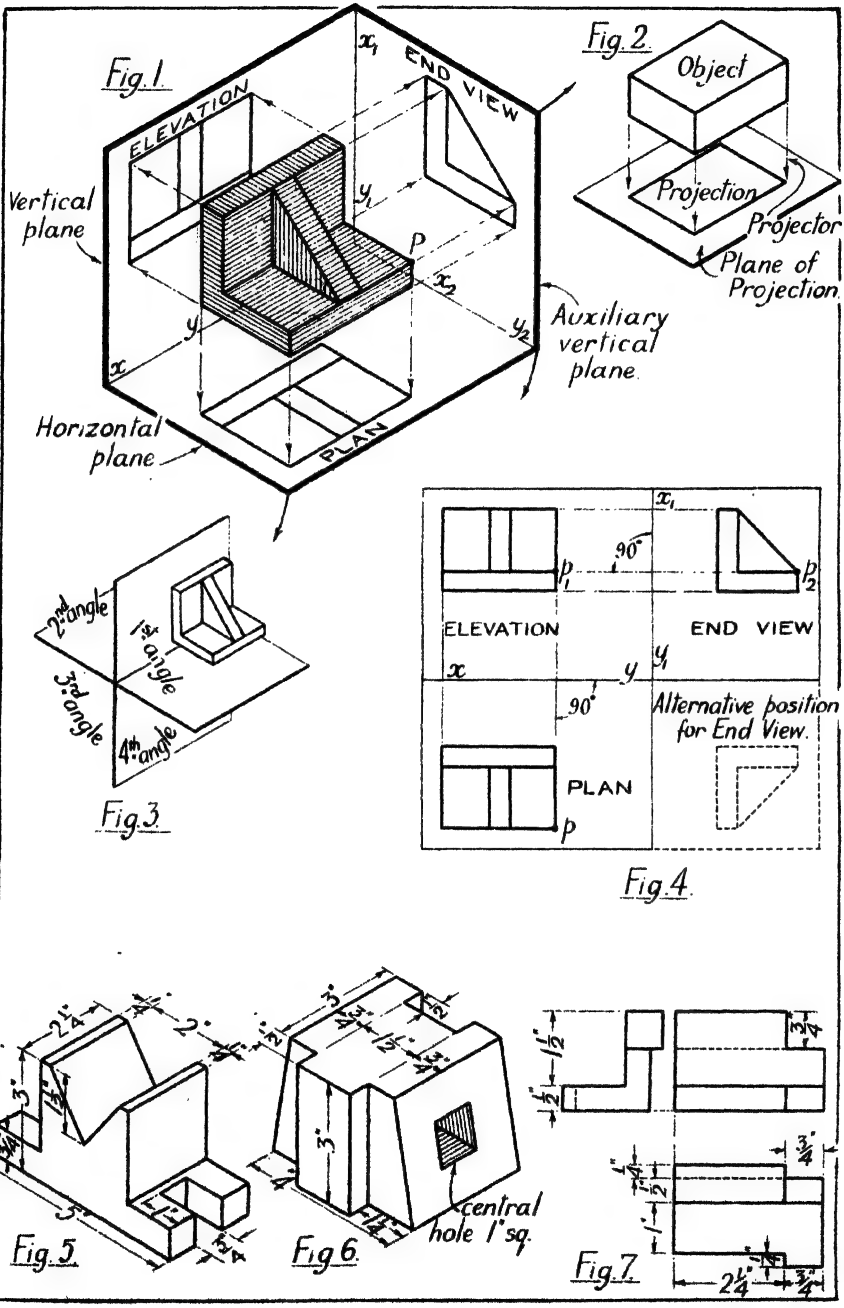



Fig. 1. Fig. 2. Fig. 3. Fig. 4. Fig. 5. Fig. 6. Fig. 7.

**3rd Angle Projection.**—Here the planes of projection come between the observer and the object and are therefore assumed to be transparent—as also are any auxiliary planes used. Projections on the planes are viewed through the planes. The student may imagine that the planes, fig. 1, are hinged together, and that after drawing on the planes what is seen by looking through them, the horizontal and auxiliary vertical planes are opened out to coincide with the vertical plane. The resulting arrangement of views is that given in fig. 2, in which the plan is *above* the elevation and the end view is *adjacent* to the end that it describes. This system is claimed by those who favour it to give an observer a clearer conception of an object than is given by 1st Angle Projection: the student can form his own opinion on the relative merits of the two systems by comparing Ex. 2 on page 8 with Ex. 3 below.

**Combination of 1st and 3rd Angle Projection.**—A compromise between 1st and 3rd Angle Projection is frequently adopted. While the plan is placed below the elevation, as in 1st Angle Projection, the end views are placed adjacent to the ends which they describe, as required by 3rd Angle Projection. This combination of systems is advantageous for drawings of long objects. As an example, the view on the end of the connecting rod partly shown in fig. 4 would be very inconveniently situated if arranged at the extreme left-hand end of the complete rod.

**Necessity for avoiding Ambiguity in Projections.** — Frequently two views only are used to describe an object, particularly if it is a detail or fragment of a larger piece and requires separate treatment. If the object is unsymmetrical and if the views are not properly labelled, it is conceivable that two dissimilar parts may be manufactured from the one sketch, according as the person interpreting the drawings is accustomed to the 1st or 3rd Angle system. Take, for example, the projections given in fig. 3: 1st Angle interpretation gives an object as at II, 3rd Angle that shown at III. The ambiguity may be avoided by inserting arrows to show the direction of aspect of the various views.

**Desirability of Familiarity with each System.**—Until there is uniformity of practice by all the important manufacturing nations, the student is advised deliberately to make himself thoroughly familiar with each system. At present, 1st Angle Projection is used on the continent of Europe and to a large extent throughout Great Britain and the Empire; on the other hand, 3rd Angle Projection is commonly used throughout North America. *Students should realize, therefore, much as they may prefer one system or the other, that both are used overseas; and that although Drawing Office practice in Great Britain may be aligned with one or the other, the fact that drawings from other countries have to be used, or the fact that designs may be called for by other countries to a specified system, leaves the student no option but to be equally at home with either 1st or 3rd Angle Projections.*

The plates in this book are largely pictorial, but the few orthographic projections used satisfy the 1st Angle system. The student will find 3rd Angle projection called for in some of the examples. In all examples, however, he should label his views to indicate clearly what they represent.

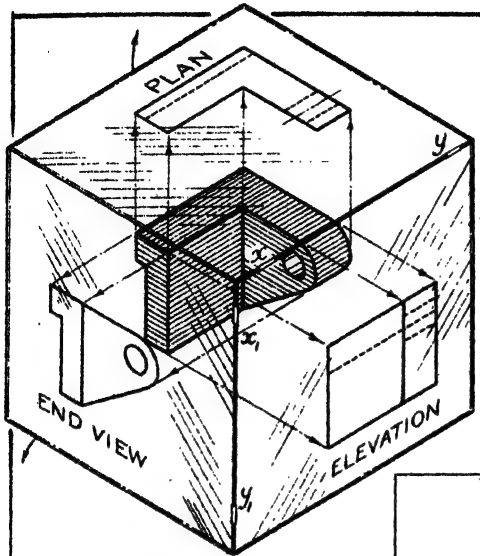
## EXERCISES

*3rd angle projection is to be used for the following*

(1) The block shown in fig. 5 has a hole $1\frac{1}{2}''$ dia. drilled centrally through it, and has a portion cut away from the lower end. The projecting boss is centrally arranged on the square face. Draw, full size, an elevation looking on the boss, a plan, and an end view.

(2) The angle block shown in fig. 6 is recessed on the hidden face and base. The recess is $\frac{1}{4}''$ deep, and a bearing strip $\frac{1}{2}''$ wide is left around the outer edges. Draw, full size, an elevation, a plan, and an end view, showing the recessed portions by dotted lines.

(3) Draw, full size, the views given in fig. 7 and project an additional end view. Then draw, freehand, a pictorial view of the object represented.



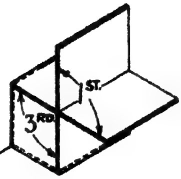



Fig. 1.

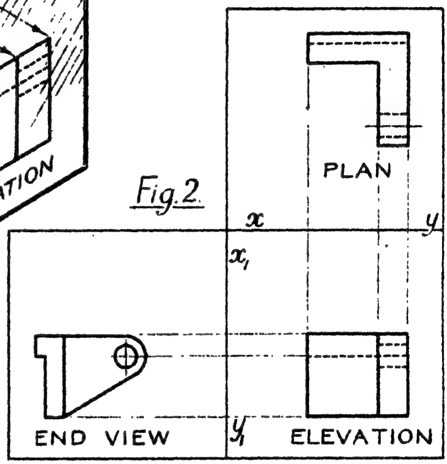



Fig. 2.

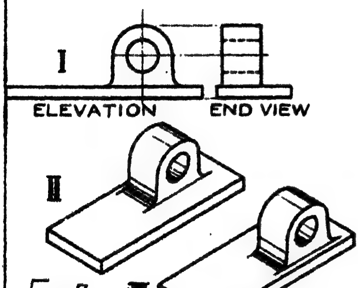



Fig. 3.

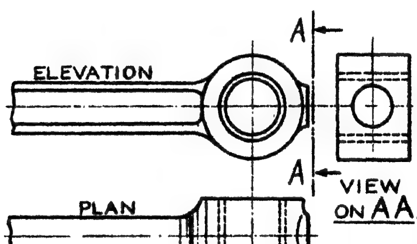



Fig. 4.

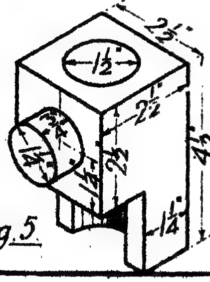



Fig. 5.

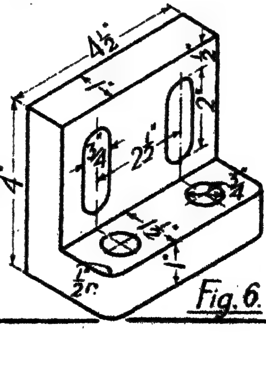



Fig. 6.

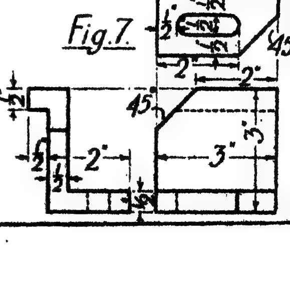



Fig. 7.

**Projection of additional views from given views.**—The majority of objects dealt with in engineering drawings can be adequately described by means of plan, elevation, and end view (or views). Occasionally, however, a projection is required on a plane taken parallel to some important face so that the true shape of the face may be shown in the projection. Such auxiliary projections are readily obtained from a given plan and elevation by using the fundamental principles of projection. The student is advised to regard the orthodox end view as a special case of the more general auxiliary projection.

**Auxiliary Projection of a Point.**—Refer to fig. 1, I. The projection of the point P on the given auxiliary plane is obtained by dropping a perpendicular from the point to the plane, $p_2$ being the projection. When both horizontal and auxiliary planes are rabatted to coincide with the vertical plane, the three projections $p$, $p_1$, $p_2$ have the positions shown in fig. 1, II. The position of the projection $p_2$ is obviously obtained thus:—draw $p_1p_2$ perpendicular to $x_1y_1$ and make the distance of $p_2$ from $x_1y_1$ = the distance of $p$ from $xy$. The projection $p_2$ is an auxiliary plan, for it has been obtained by projection from the elevation $p_1$. In a similar manner the projection of a point on any other auxiliary plane may be obtained from a given plan and elevation. The method is important and is based upon the following fundamental rules:—

(1) The projections of a point are on a straight line perpendicular to the ground line.

(2) The distances of all plans (or elevations) of the same point from the corresponding ground lines are equal.

**Auxiliary Projection of a Solid.**—By applying the construction of fig. 1, II, to selected points on the outline of the solid, auxiliary projections may be obtained almost mechanically, providing the given plan and elevation of the solid show also the plan and elevation of the points chosen: for although the plan and elevation of a line may be given, the corresponding projections of a point in the line may not be at once available—as an example, in the line $ab$, $a_1b_1$, fig. 4, the plan corresponding to the elevation $p_1$ of a point in the line could not be marked by inspection.

In fig. 2, I, the solid shown has been projected point by point on to the given auxiliary plane. The rabatted view given in fig. 2, II, is readily obtained by projecting the various corners of the solid and joining them in the correct order. The view is an auxiliary plan.

When $x_1y_1$ is perpendicular to $xy$, the auxiliary plan becomes the ordinary end view. *Hence the projection of an end view from a given plan and elevation admits of no ambiguity if the principles discussed above are applied rigorously.* The examples given below are to be treated in this way.

## EXERCISES

*All to be drawn full size*

(1) Draw the two views given in fig. 3 and project a plan. Regard the left-hand view as the elevation. Note: the views admit of various interpretations; show the simplest object.

(2) Draw the elevation and plan given in fig. 4 and project the end view given when looking from left to right.

(3) A plan and elevation of a regular hexagonal prism is given in fig. 5. Project an auxiliary plan on $x_1y_1$. Then ignore the original plan, and using the original elevation and the auxiliary plan, project an end view from the left, *along* $x_1y_1$.

(4) Draw the views given in fig. 6 and project (*a*) an end view from left to right, (*b*) an auxiliary plan on $x_1y_1$ (i.e. a view on the underside).

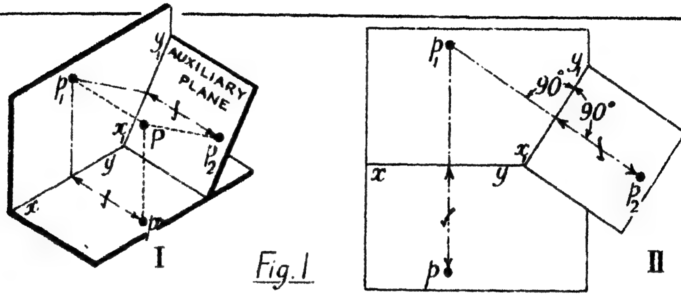



Fig. 1

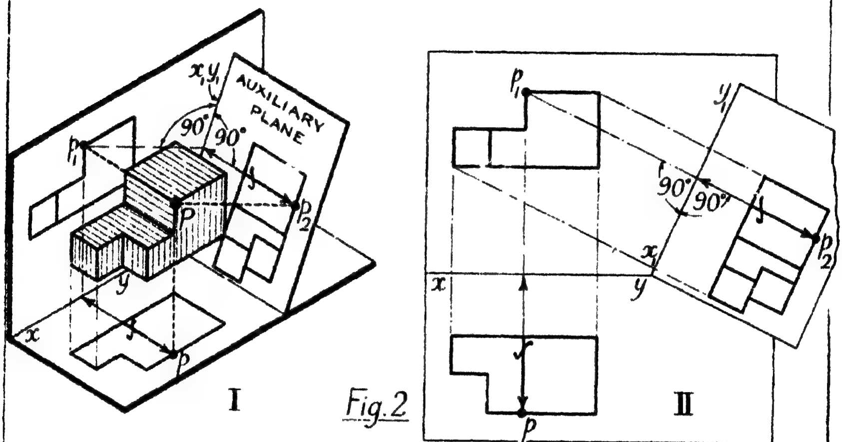



Fig. 2

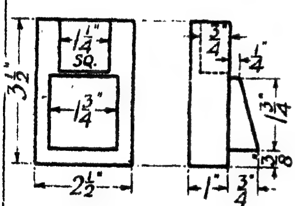



Fig. 3

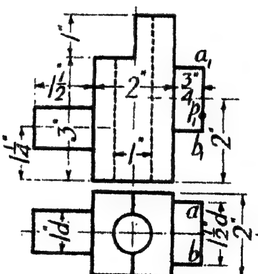



Fig. 4

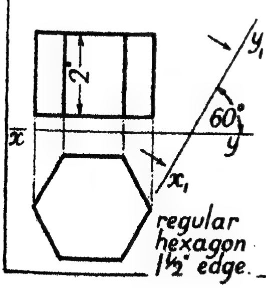



Fig. 5

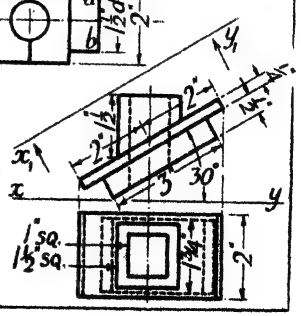



Fig. 6

**Sectional Views.**—These are employed to reveal the shape or construction of parts not directly visible, the indication of which by dotted lines would be either inconvenient or confusing.

In preparing sectional views we imagine the object to be cut by planes, so that the portion of the object lying between the observer and the planes may be removed to reveal some particular part previously hidden from view. The following examples illustrate the method:—

(1) Bearing Step, fig. 1, I. The shape of the body of the step may be shown by taking a section plane as at I and removing both the plane and the piece to the front of it, as at II. The surface actually cut is shown by fine sloping lines—called cross-hatching or section lines.

(2) Slide Valve, fig. 2, I. The shape of the cavity is best shown by sectional views. Those given are obtained by dividing the valve symmetrically by planes perpendicular to the lower face—longitudinally through the valve as at II, and transversely as at III.

**Conventional Treatment of Sectional Views** (*see also pages* 16–19).—The rules of orthographic projection apply to sectional views, which take the position of the external views they replace. The trace * of the imaginary cutting plane is indicated by a thin long chain line. Where necessary the section plane is lettered and arrows are inserted to give the direction of view, as in fig. 2, II, page 17.

When a sectional view replaces an exterior view, all outlines visible beyond the plane of section must be shown, as in fig. 1, III; otherwise it is sufficient to show only the part sectioned.

A common error with beginners is the omission of the edges of a hole, beyond a section plane, when the plane passes through the hole. The student should imagine his set square to be cut through the hole by a section plane, as in fig. 3: the correct sectional view is shown.

Sections in metal are invariably cross-hatched by fine lines evenly spaced, extending only over the areas cut by the section plane. The lines should be spaced by eye and should always slope at 45°. The spacing is governed mainly by the extent of the area to be sectioned. If the lines are too closely spaced the operation of sectioning becomes tedious and lengthy; for the examples herein a suitable spacing is from $\frac{1}{8}''$ to $\frac{3}{16}''$ with a closer spacing of about $\frac{1}{16}''$ for bushes and other small parts.

*Note.*—The appearance of a drawing is often marred by badly spaced overheavy section lines.

## EXERCISES

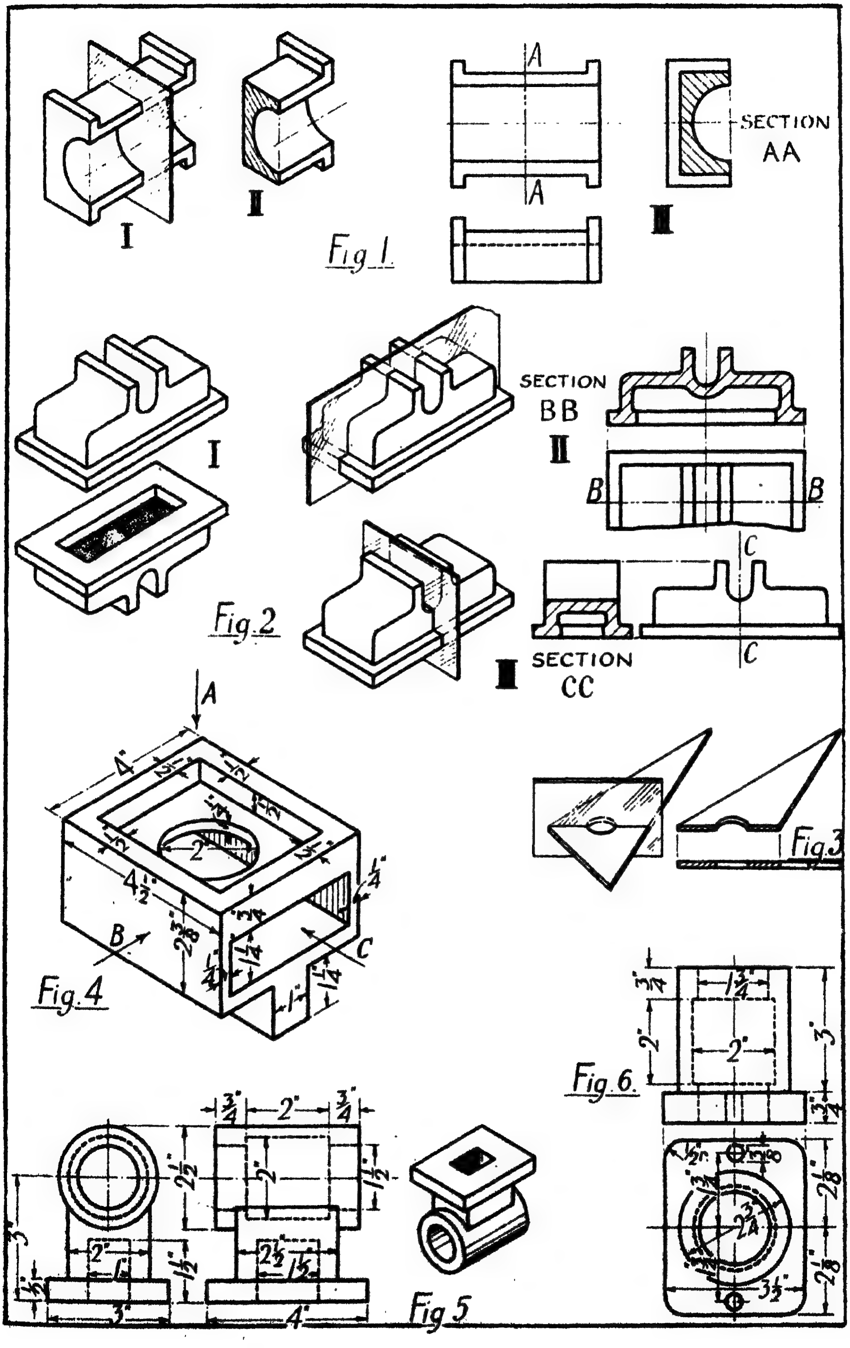
*All exercises to be drawn full size. Spacing for section lines, about* $\frac{1}{8}''$

(1) Fig. 4.—Draw a plan and elevation of the given object in the direction of the arrows A and B and project a central sectional end view in the direction of the arrow C.

(2) Fig. 6.—Draw the two views given and add a sectional end elevation taken through the bolt holes.

(3) Fig. 5.—Draw the two views given. Project a plan and a sectional end view.

* I.e. the line of intersection of the cutting plane and the plane of reference.


I
II
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A
SECTION AA
III
Fig. 1.
SECTION BB
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B
B
C
C
Fig. 2
III
SECTION CC
A
4"
1/2"
1/2"
1/4"
1/2"
2"
1/2"
1/4"
4 1/2"
1/4"
3/4"
2 3/8"
B
1 1/4"
C
1/4"
1 1/4"
1"
Fig. 4
Fig. 3
3/4"
1 3/4"
2"
2"
3"
Fig. 6.
3/4"
1/2 r.
1/4 Dr.
1 3/4"
2 3/4"
2 1/8"
2 1/8"
3 1/2"
3/4"
2"
3/4"
2 1/2"
2"
1 1/2"
3"
2"
1 1/2"
1 1/2"
2 1/2"
1 1/2"
1/2"
3"
4"
Fig 5

**Treatment of Composite Objects.**—When a composite object is sectioned, the various parts are distinguished in the sectional view by a change in the direction, but not in the inclination, of the section lines. This is illustrated in fig. 1, II, which shows a cross section of the slide in fig. 1, I. When it is impossible to avoid the same direction for section lines in adjacent parts, a distinction may be made by varying the spacing of the lines. No variation should be made in the inclination of the section lines, this being always 45°.

**Half Sections.**—When an object is symmetrical in form its construction may be shown more clearly by giving an exterior view on one side of a centre line and a sectional view on the other, as in fig. 1, III. This method is widely adopted to avoid either the construction of an additional view or the excessive use of dotted lines. Quarter sections or fragmentary sections may be used where half or full sections are unnecessary, as shown herein.

**Staggered Sections.**—These are used where it is desired to show, in one sectional view, parts which are not in the same plane. An example is given in fig. 2. To show both branches of the object in one sectional view, planes are taken as at I, to give the section shown at II. It should be noted that the lines which would appear in the section to indicate the corners produced by the change of planes are omitted from the view. The traces of the section planes are indicated at XX in the plan.

**Conventions for Materials.**—The cross-hatching adopted to represent materials in common use is shown in fig. 3. It is not customary to rely solely on these conventions, and when preparing a working drawing the materials for the various parts should be indicated either by notes or by a key diagram (see page 33).

When drawings are to be coloured, sectioned parts are tinted more deeply than external views. The following are the colours generally adopted:—

Cast Iron—Payne's Grey.
Wrought Iron—Prussian Blue.
Steel—Purple (Prussian Blue and Crimson Lake).
Brass and Gunmetal—Gamboge.
Copper—Crimson Lake with Gamboge.
Wood—Burnt Sienna.
Brickwork—Light Crimson Lake.
Concrete and Stone—Light wash of Gamboge.

**Interpretation of Sectional Views.**—Isolated sectional views are liable to be misinterpreted, and the student should endeavour always to visualize the complete part. Failure to do this may result in such grotesque proposals as that illustrated in fig. 4—a form of wheel rim to prevent derailing of rolling stock!

## EXERCISES

(1) A part section of a casting and nozzle piece is shown in fig. 5, all horizontal sections being circular. Draw, full size, the half view given to the left of the centre line, add the complementary *outside* half view, and project a plan.

(2) Fig. 6 shows a gland suitable for a small pump rod. The gland is fitted with a bush which passes through to the lower face of the elliptical flange. The width of the flange is 2½". Draw, full size, the following views: elevation, half in section; sectional end view; plan. Show the gland with the flange uppermost.

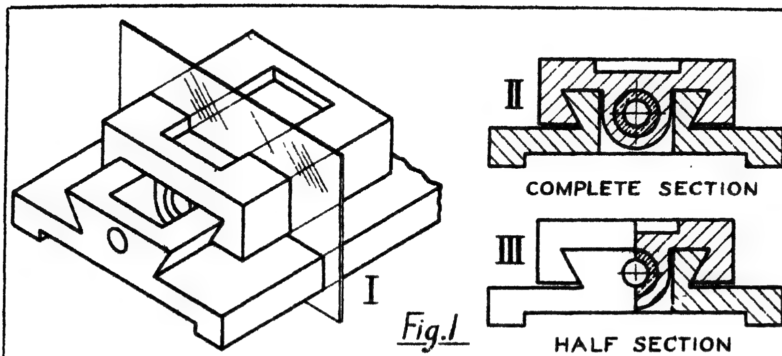



Fig.1

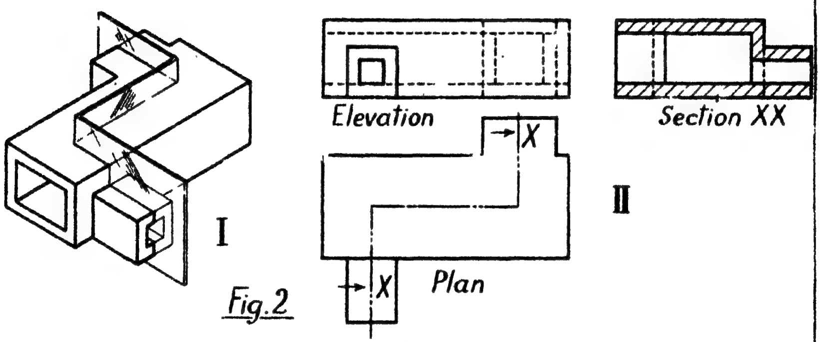



Fig.2

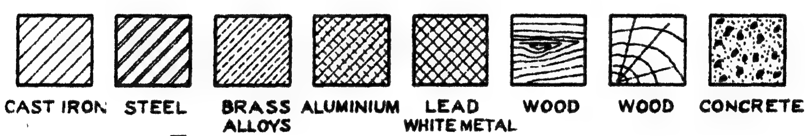



Fig 3. CONVENTIONS FOR MATERIALS

Fig.4.

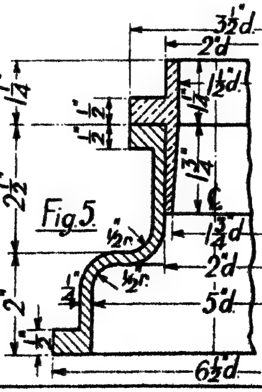



Fig.5.

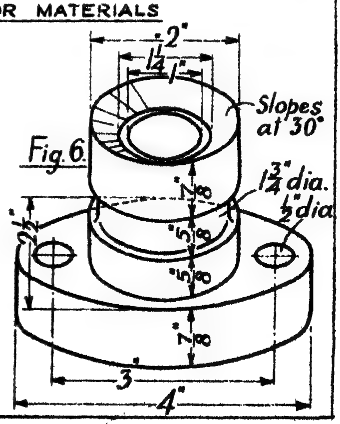


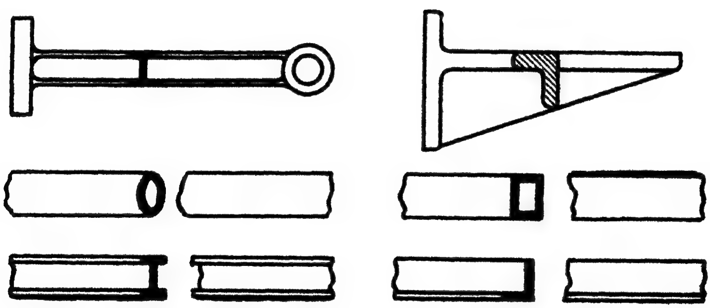

Fig.6.

**Conventional Treatment of Webs and Ribs.**—Thin webs and ribs are largely used to strengthen castings, as in fig. 1. When a section plane passes through a web longitudinally, the section is treated as shown in fig. 1, II, not as at I, in order to avoid the appearance of solidity. If for any reason it is essential to indicate that the section plane does pass through the web, the treatment at III may be adopted. Webs and ribs are sectioned in the ordinary way if the section plane cuts them transversely.

**Parts not Sectioned.**—Many fittings which appear in sectional drawings of machine parts are better shown by external views, and it is conventional practice *not to section* the following parts when their axes or long dimensions lie in the plane of section (or are parallel to it):—shafts (unless very large); bolts; screws; nuts; rivets; keys; cotters; pins and small cylindrical parts. The composite drawing, fig. 2, illustrates the treatment of these parts in a sectional view: it should be noted that the shaft is only partly sectioned to reveal the key and the cotter.

In addition to the foregoing, it is unusual to section the arms of valve wheels, pulleys, or flywheels, when the axes of the arms lie in the plane of section; nor are the teeth of gear wheels sectioned (see later illustrations).

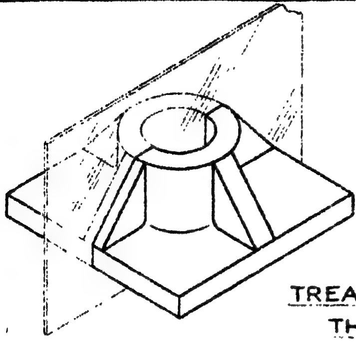
**Revolved Sections.**—These are often used for rods, brackets, &c., to avoid the construction of the orthodox sectional view. Long bars may be shown broken and the shape of the section indicated in freehand. Examples illustrating the use of revolved sections are given in the figure.



## EXERCISES

(1) The object shown in fig. 3 consists of a cylinder, on an equilateral triangular base, supported by *three* webs. The corners of the base are rounded off and the hole passes centrally through both cylinder and base. Draw a sectional elevation, the plane of section passing through the web A; an end view in the direction B; and a plan.

(2) Draw an elevation of the bracket, fig. 4, in the direction of the arrow C; a sectional end view, in the direction of the arrow D; and an underneath view.



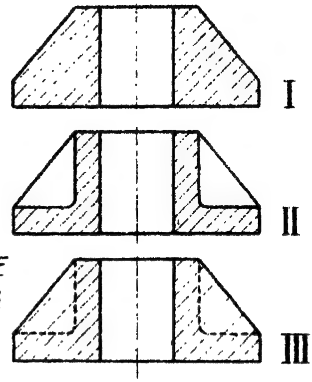
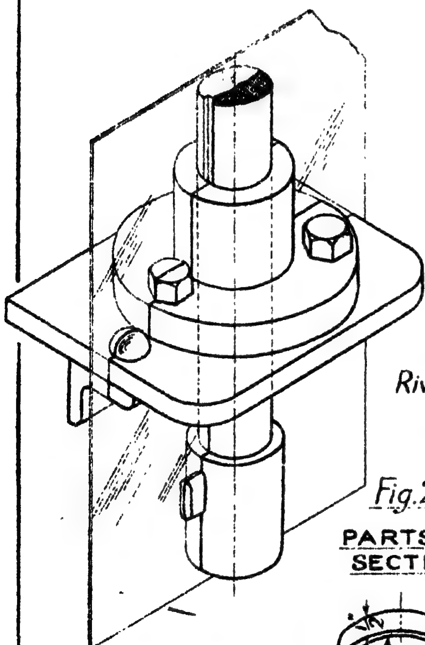



Fig. 1.
TREATMENT OF THIN WEBS



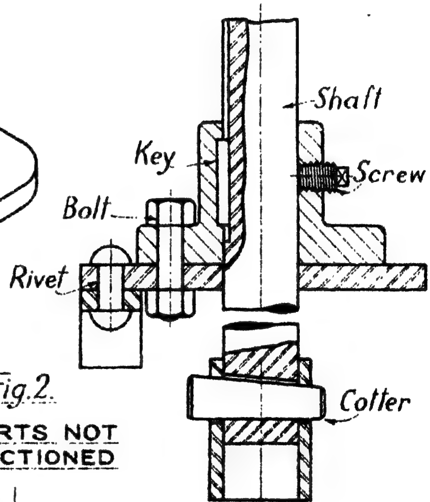



Fig. 2.
PARTS NOT SECTIONED

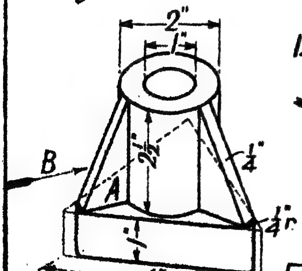



Fig. 3.

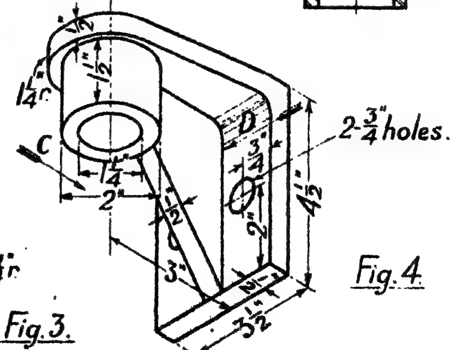



Fig. 4.

**Lettering.**—Particular care should be given to the formation, spacing, and alignment of letters in the notes on a drawing. Indifferently formed lettering is at once noticeable—the eye being accustomed to perfection in the type on a printed page—and is sufficient to spoil the appearance of an otherwise good drawing.

Titles or main headings should be printed (or stencilled) in block letters. Sub-headings and notes may be printed in script of a simplified form lending itself to rapid production, as used on the plates herein. Guide lines should be drawn for all lettering: lines about ⅛″ apart will give printing large enough for most purposes, the letters being ***extended rather than heightened*** where a ***bolder*** appearance is desired.

**Dimensioning** (*see also page* 22).—The correct dimensioning of a drawing calls for considerable thought, and although rules may be given for general guidance, proficiency is reached only after experience both in the preparation and interpretation of working drawings.

As shown in fig. 5, both dimension and extension lines should be thin and continuous, the latter being broken at the outline of the views. Dimension figures should be placed in reasonable positions and so arranged that they are not crossed by any other line. Centre lines should not be used as dimension lines. Arrow heads should be a full ⅛″ long and partly filled in. The following conventions should be adopted.

(1) Dimension figures to stand normal to the dimension lines when read from the base or right-hand side of the drawing, and to be inserted in a break in the dimension line. The method is illustrated in fig. 1 (which, however, is not *completely* dimensioned); the arrangement of oblique dimensions should be noted.

(2) All vulgar fractions to be written with the line dividing the figures parallel to the dimension line.*

(3) The decimal point should be bold, and opposite the middle of the figure.

(4) All dimensions should be direct, i.e. they should not involve calculations.

(5) The diameter of a complete circle should be given in preference to the radius. Diameters of pitch circles should be followed by the letters P.C. or P.C.D. If a circular part appears in one view only, but not as a circle, the word "dia." should follow the dimension, as in fig. 2 (for a square part add "sq."). Various ways of dimensioning diameters and radii are shown in figs. 2 and 3.

NOTE.—With certain exceptions, e.g. cylinder bores, dimensions up to 24″ are usually given in inches, and those above 24″ in feet and inches. Where drawings are dimensioned wholly in one unit, the symbols in. or mm. may be omitted provided a note is added stating the units for the drawing.

(6) A taper should be defined as unit alteration in diameter or thickness of a part in a specified length, the latter being measured along the geometric centre line. Methods of indicating tapers are given in fig. 4. When the direction of a taper is not evident from the drawing, it should be shown by a wedge on the centre line.

(7) Where a dimension on a scale drawing does not agree with the scaled distance owing to changes subsequent to the completion of the drawing, it should be underlined heavily or marked N.T.S.

Many of the above are included in fig. 5.

**Exercise.**—Draw, twice full size, the views given in fig. 5 and project a sectional plan. Without referring to the book insert all necessary dimensions, scaling them from your drawing; then compare the result with that given.

* This has not been wholly adhered to herein, e.g. the inclined division line has been used frequently to save space.

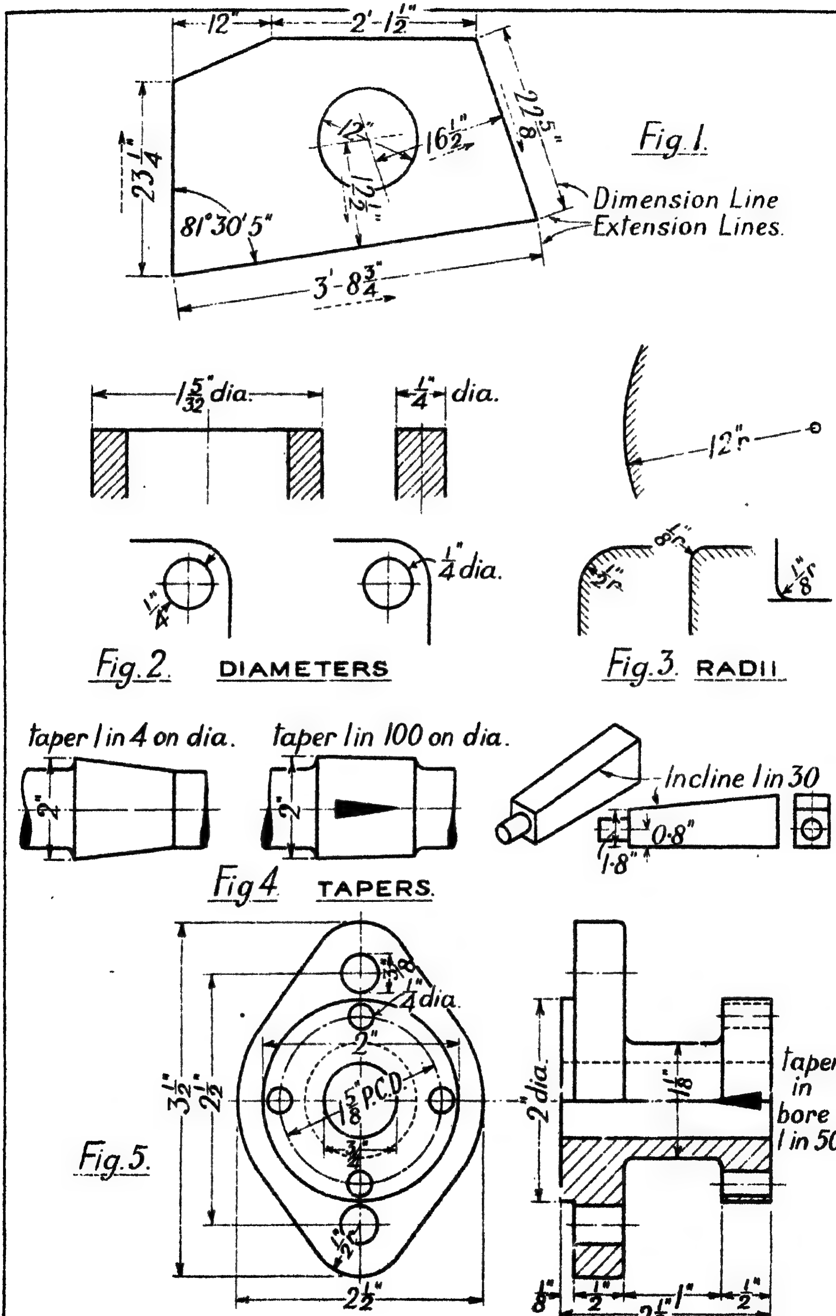



Fig. 1.

Fig. 2. DIAMETERS

Fig. 3. RADII

Fig. 4. TAPERS.

Fig. 5.

**Location and Size Dimensions.**—A machine part may be regarded as a connected group of simple geometric solids. It is convenient to distinguish between dimensions which specify the size of the separate solids, and those which locate the relative positions of these solids. For the bracket shown in fig. 1, the **location dimensions (L)** are given in fig. 3, and the **size dimensions (S)** in fig. 4: the two figures should be closely compared. Location dimensions for cylinders and cones should be taken from the axes, not from the curved surfaces of the solids.

When an object is to be *partly* machined, as is often the case with castings and forgings, it is essential that the location dimensions for the machined surfaces should be referred to some axis of symmetry or finished plane surface; not to an edge or surface on the unmachined part. The surfaces to be machined on the bracket in fig. 1 are clearly indicated, and the corresponding edges are marked *f* (signifying finish)* in the projections given in fig. 2. All the location dimensions, fig. 3, are taken from machined parts.

If an object is not to be machined, or if no indication is to be given on the drawing as to which surfaces are to be machined, the student must select suitable surfaces or axes of reference for the location dimensions.

**Systems of Dimensioning.**—Dimensions may be arranged in several ways, each having some advantage over the others. Three systems are illustrated opposite:—

(1) Dimensions *wholly outside* the views, as in fig. 5. The dimensions need not be cramped and may be made prominent. (2) Dimensions *wholly within* the views, as in fig. 6. Comparatively few extension lines are required and the outline of the object is shown without interference. (3) Dimensions *partly within and partly without* the views, as in fig. 7. By judicious arrangement the advantages of (1) and (2) may be combined.

Although no definite rule can be given, in general the first system should be adopted.† The shortest dimension lines should always be arranged nearest the view.

**Relative Importance of Dimensions.**—A dimensioned drawing of an article should be capable of one interpretation only. It should result in the production of exactly similar articles. This is an ideal which is not easily achieved; indeed, strictly speaking, it is unattainable, as the following considerations show. A dimension given as 2″ really means that the size may lie on either side of 2″ within the customary limits of manufacture. By accurate working, the resulting size may lie somewhere between 1·995″ and 2·005″. Hence, the dimensions of one article may differ from those of another made from the same drawing; and, although the differences may be reduced, exact similarity will be always relative and not absolute. It is true that for a great many articles the small size differences inevitable in manufacture are unimportant; but it is also true that for many others, particularly where interchangeability is to be secured, the differences must be held within closely prescribed limits. This requirement is indicated by replacing a simple dimension by two dimensions within which the size must be kept. For example, 2″ would be replaced by $\frac{2{\cdot}003}{1{\cdot}997}$ $\left(\text{or by } 1{\cdot}997 \begin{smallmatrix}+\\-\end{smallmatrix}\begin{smallmatrix}\cdot006\\0\end{smallmatrix}\right)$.

The settlement of these limiting dimensions, and their insertion on a drawing, are fully discussed later—see pages 160 to 167.

**Exercises.**—The objects shown in figs. 8, 9, and 10 are to be partly machined, as indicated: the darkened surfaces are to be left rough. Draw *freehand* the necessary orthographic views of each object and insert location and size dimensions as in figs. 3 and 4.

* See also p. 32.

† In certain classes of work, e.g. plate metal work, dimensioning is carried out from two axes of reference, usually at right angles to each other.

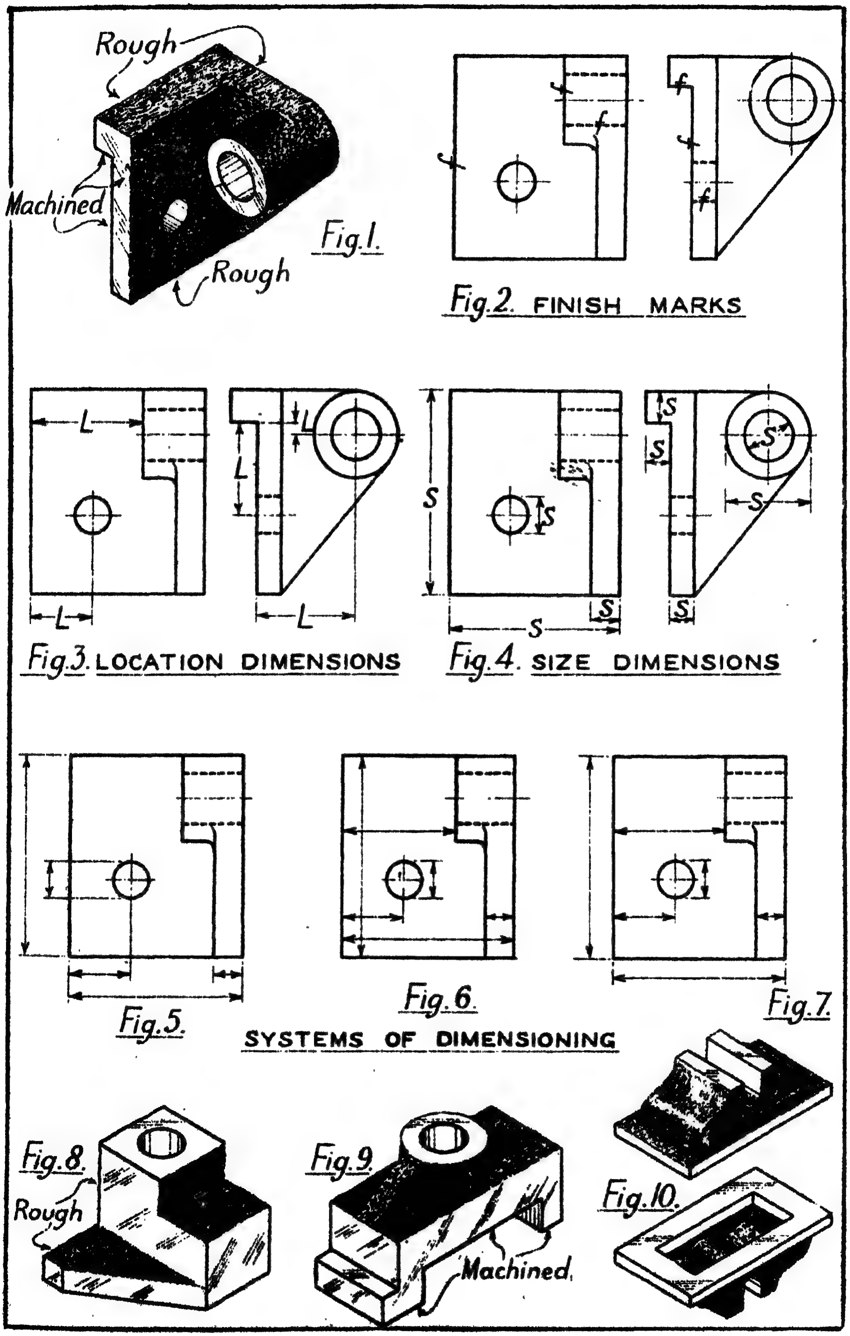



Fig. 1.

Fig. 2. FINISH MARKS

Fig. 3. LOCATION DIMENSIONS

Fig. 4. SIZE DIMENSIONS

Fig. 5. Fig. 6. Fig. 7.

SYSTEMS OF DIMENSIONING

Fig. 8. Fig. 9. Fig. 10.

**General.**—The majority of machine parts have their principal axes or centre lines in three directions at right angles to one another, i.e. in the directions of the concurrent edges of a cube. For such as these, a system of pictorial projection, called isometric projection, is particularly suitable. It has been largely used throughout this book and its advantages are evident. Space will not permit of a discussion of the principles upon which the system is based; all that can be given here is a brief description of a method of preparing isometric drawings.

Isometric projections are often avoided because of the labour involved in plotting the ellipses which, in these views, represent the outlines of circular parts. It will be shown, however, that by the aid of an isometric scale and a pair of proportional compasses the process may be simplified considerably and the confusion usually associated with these views avoided.

**Projection of a Rectangular Solid.**—Fig. 1 shows the plan of a cube when a long diagonal of the solid is vertical: the outline is a regular hexagon. As arranged, lines representing the edges of the cube are either vertical or inclined at 30° to the horizontal. The edges of the cube which meet at the upper and lower corners are equally inclined to the horizontal plane, and their plans (which are concurrent lines meeting at 120°) are therefore shortened to the same extent. If we determine the amount of this shortening, we may project any rectangular solid isometrically, for its edges will be parallel to the three lines OA, OB, and OC—called the ***isometric axes***—and shortened proportionately. Fig. 2 shows a column base drawn in this way. The scale for measurements along the axes is now discussed.

**Isometric Scale.**—For the method described here and overleaf, a scale is required for all measurements along the axes. Refer again to fig. 1. Join AC and draw AE and CE at 45° to AC. The $\Delta$AEC represents the true shape of the $\Delta$ADC, and either AE or EC gives the actual length of the edges AD or DC. This is the basis of the isometric scale. All distances set off along the axes OA, OB, or OC should be made shorter in the ratio AD/AE.

Refer to fig. 3.—Set off AE at 45° and AD at 30° to a base line AC. Graduate AE in inches and parts of an inch, and from each point along AE draw perpendiculars to AC, thus dividing AD in a similar way. The graduations along AD are those for the isometric scale. Transfer these divisions to the edge of a bevelled plain ruler as indicated (mark them with a knife) and *use this scale ruler for all measurements along the isometric axes.*

**Isometric Projections of Solids not wholly Rectangular.**—One method of treating these solids is shown in fig. 4, which gives the isometric projection of a hexagonal prism. The prism is assumed to be enclosed in a rectangular frame, so that the edges of the solid may be located by reference to the frame. The points lettered in the figure can be readily obtained on the isometric frame, and the figure completed. It is usually necessary to draw a portion of some orthographic view of the solid, from which view distances may be transferred to the isometric view. Either the orthographic views may be drawn to the isometric scale and the distances transferred directly, as has been done in fig. 4, or they may be drawn to a natural scale and distances transferred first to the isometric scale and then to the drawing.

**Exercises.**—Draw isometric views of the following figures: p. 9, figs. 5, 6, and 7; p. 13, figs. 3 and 6; p. 15, fig. 4 (omit hole).

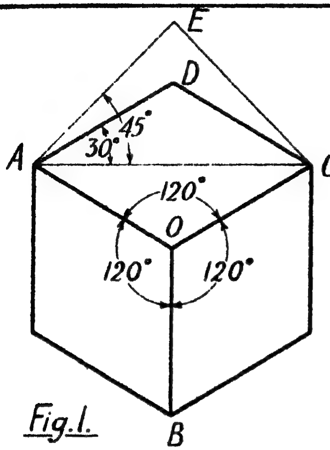



Fig. 1.

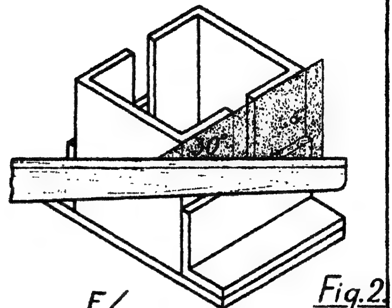



Fig. 2.

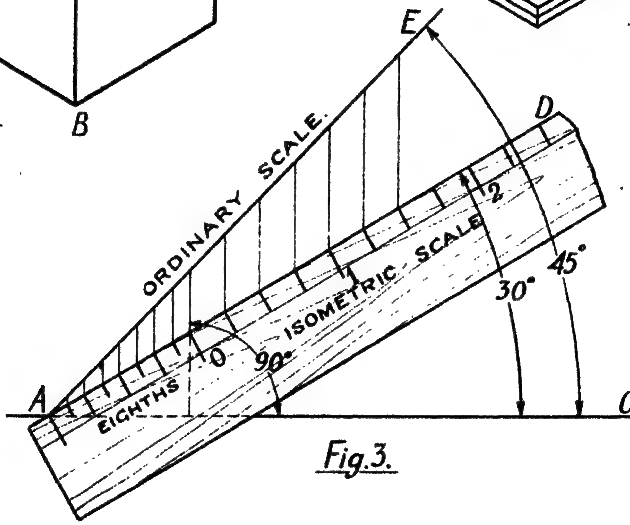



Fig. 3.

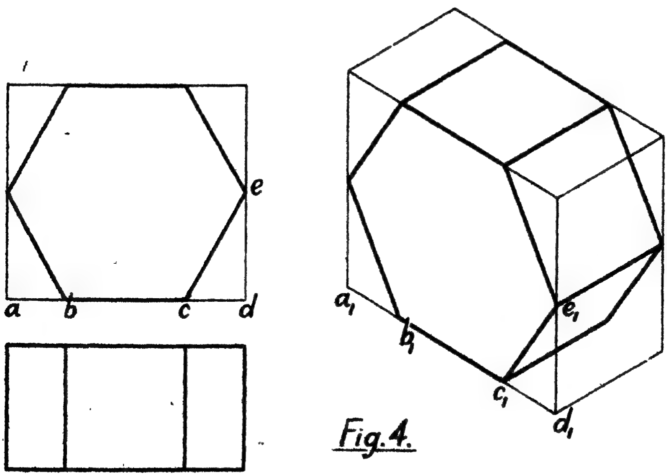



Fig. 4.

**Projection of Circles on Isometric Planes.**—Any plane parallel to two of the isometric axes is called an *isometric plane.* In fig. 1 the faces of the cube are taken as isometric planes: they have been lettered X, Y, and Z for purposes of reference.

The isometric projection of a circle is always an ellipse, with major axis equal in length to the diameter of the circle. Circles drawn on the faces of the cube will appear as in fig. 1. When a machine part is being projected isometrically it can usually be arranged so that all circular faces lie in planes parallel to the planes X, Y, and Z; hence the ellipses in the majority of isometric views will be similarly situated to those in fig. 1.* A method of drawing these curves will now be given which requires few construction lines and is simple and speedy.

**Method.**—Let it be required to draw the isometric projection of a circle of radius R on planes X, Y, and Z, fig. 1, having its centre at C. In the following method five points on each ellipse are located: one at each end of the major axis, one at each end of the minor axis, and one other. An approximate ellipse is then drawn through these points, using compasses. The construction is shown separately and in steps, for each plane, in figs. 2, 3, and 4; except where otherwise mentioned, the description given applies to each figure.

Refer to figs. 1, 2, 3, and 4.—As stated above, the length of the major axis AB = 2R; its direction for plane X is horizontal, for planes Y and Z inclined at 60° to the horizontal. The length of the minor axis = 0·58 × major axis, very nearly.† The major and minor axes bisect each other at right angles. Set the index of a pair of proportional compasses to read 0·58 and retain this setting. Proceed as follows, using tee square and 30° set square.

**Construction.**—Through C draw the major axis AB, horizontally for plane X (fig. 2), inclined at 60° to the horizontal for planes Y and Z (figs. 3 and 4). Set the large end of the proportional compasses to R, and mark off CA and CB = R. Without altering the compasses, set off the small end span along CD and CE drawn at right angles to AB, giving D and E. Along CF, making 30° with AB, set off the radius of the circle *on the isometric scale* (for planes Y and Z, CF is vertical). With centre along BA draw, by trial, a circular arc to pass through B and F: draw a similar arc through A. With centre along ED produced draw a circular arc to pass through E and to touch tangentially the first arc: draw a similar arc through D. This completes a figure which approximates closely to the correct ellipse. The ellipses for the plates in this book were set out in this way.

**Applications.**—The method is particularly suitable for the isometric projection of objects having many circular parts lying closely together. Fig. 5 shows a simple machine part drawn in this way. *All the construction lines necessary are shown.* The circular outlines have been taken in planes parallel to plane X (fig. 1), but they might equally well have been taken in plane Z, or in plane Y as in fig. 6. The dotted interior lines in fig. 5 may be avoided by sectioning the object along lines parallel to the isometric axes, as in fig. 6: this method has much to recommend it and has been largely adopted herein.

**Exercises.** (1) Taking dimensions from fig. 6, draw the view given in fig. 5, but show the right-hand front quarter of the object removed.

(2) Draw isometric views of the following objects: p. 11, figs. 5 and 6; p. 13, fig. 4; p. 15, fig. 5; p. 19, fig. 4; p. 21, fig. 5.

* The isometric projection of irregular objects and axometric projection are discussed in the author's *Practical Geometry and Engineering Graphics.*

† Actually = major axis ÷ $\sqrt{3}$.

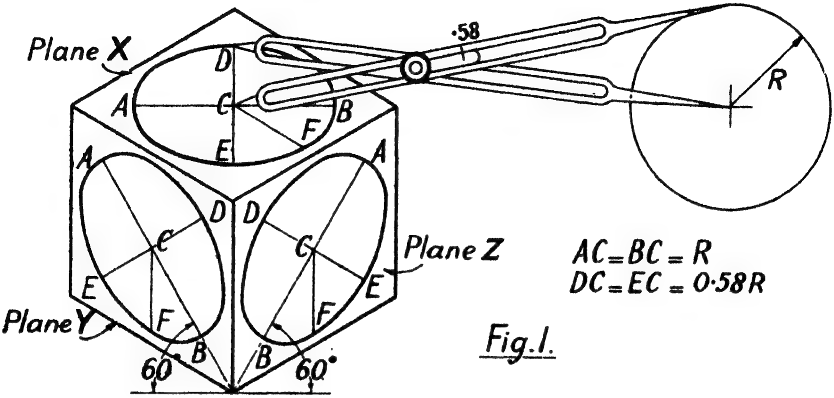



Fig. 1.

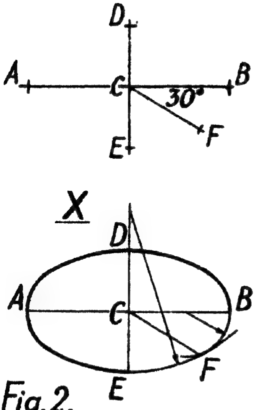



Fig. 2.

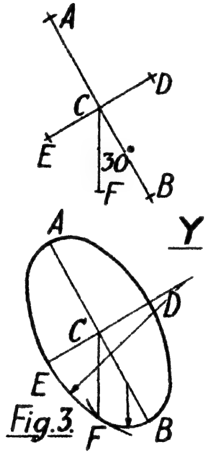



Fig. 3.

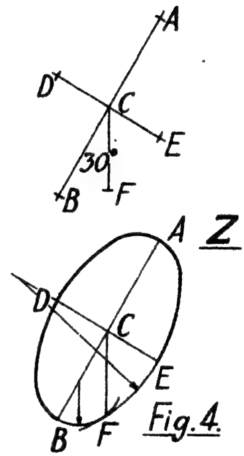



Fig. 4.

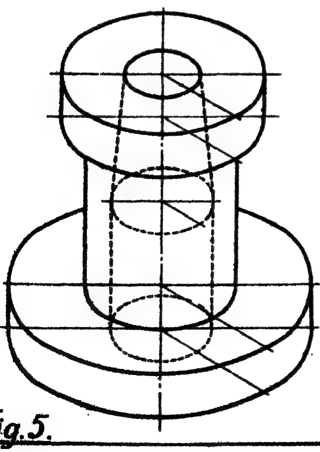


Fig. 5.

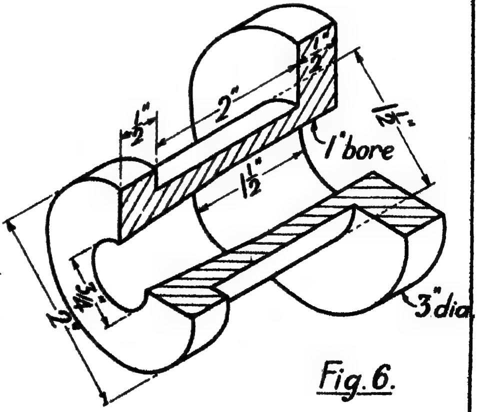



Fig. 6.

**General.**—The distorted appearance of isometric views may be largely avoided by the use of perspective projection, which, although commonly used by architects and artists, has hitherto been less used by engineers, mainly because of the labour involved. This has been much reduced by various means, and perspective is now more in favour in engineering drawing offices. Only a brief account of the method can be given here.*

In perspective, the projectors converge to a point, and the required view is given by an imaginary plane inserted to cut the converging lines. Fig. 1 shows: a cube in plan and elevation; an inclined plane, traces VTH; and a point of sight, projections $s$, $s_1$. Rays from the corners of the cube are taken to $s$, $s_1$; their intersections with VTH give a perspective view of the cube. The method is illustrated by the projection of two points only, F and G (projections $f$, $f_1$, and $g$, $g_1$), all essential construction lines being shown. The completed cube, with its central axes, A, B and C, is drawn to a larger scale in fig. 2.

**Perspective Scales.**—The cube in fig. 2 differs from an isometric view in that the edges all have different inclinations and lengths. Hence it is not possible to use three concurrent edges as axes nor to have a common scale. An alternative is to refer to the ***central axes***, suitably scaled, and to make use of ***vanishing points*** to which parallel edges converge (see fig. 4). If $V_A$ is taken infinitely distant, *vertical* edges are parallel; the result is the kind of view favoured by architects.

The projection of the ***perspective scales***, the divisions of which are not uniform, is merely a repetition of the method of projection for F and G in fig. 1, taking equally spaced points on the central axes. The result is shown in fig. 2, which gives inch scales, about half full size.

There is no limit to the variety of views of the cube, and hence of dispositions of the axes, depending upon the positions of the plane VTH, and the viewpoint $s$, $s_1$. It is necessary then to select a few standard positions, and to determine for them the axes, scales, and vanishing points, for use in routine work.

**Making a Perspective View.**—Fig. 3 shows orthodox views of a simple bracket to a scale of ½. A perspective view of this bracket is shown in fig. 5 to a scale of ½. The positions of the three axes, dotted fig. 3, are settled arbitrarily. The location of points in the perspective view is illustrated by the treatment of the twin point P on the curved edge. The distances $a$, $b$ and $c$, in fig. 3, are taken along the scales A, B and C in fig. 5, and the points P fixed by projection lines *drawn in perspective*, i.e. they converge to vanishing points (which are not shown). Fewer construction lines may in fact be used. Alternative views may be drawn by making other axes vertical—i.e. by turning fig. 2 into five other positions.

**Mechanical Aids.**—The use of distant vanishing points is not always convenient, and time can be saved by using a drawing-board cut along arcs struck from the vanishing points, as in fig. 4, and by using a tee-square having a stock giving two-point contact on the arc. For much routine work this combination has obvious advantages.

## EXERCISES

(1) Taking a cube 2½″ edge, a plane VTH, and point $s$, $s_1$, placed as in fig. 1, make a perspective view of the cube and obtain the vanishing points and scales. $s$ is centrally placed in plan.

*Answer.*—(Fig. 4) ∠COA = 61°; ∠BOA = 50°; $OV_A = 11''$; $OV_B = 9''$; $OV_C = 12''$, approx.

(2) Using the data of Ex. 1 construct perspective views of ($a$) the bracket in fig. 3, and ($b$), the brackets on pages 169 and 202.

* For fuller details the student is referred to the author's *Practical Geometry and Engineering Graphics.*

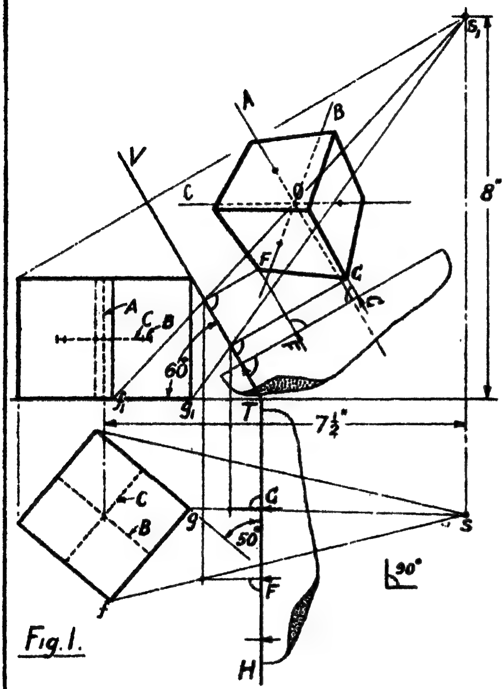



Fig. 1.

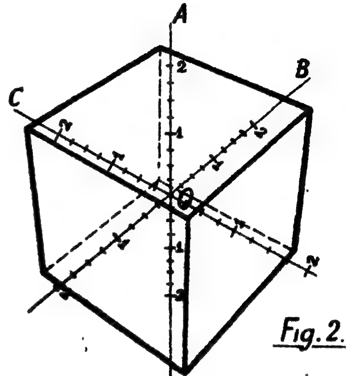



Fig. 2.

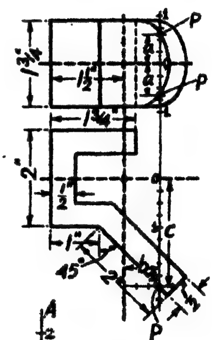



Fig. 3.

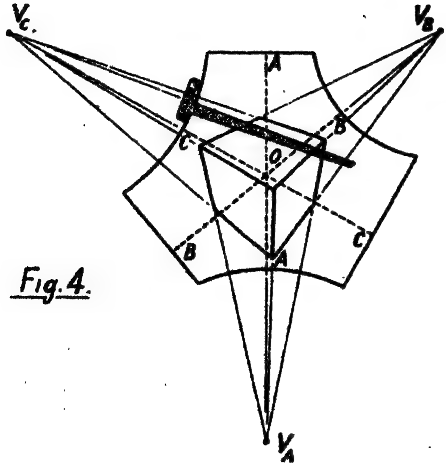



Fig. 4.

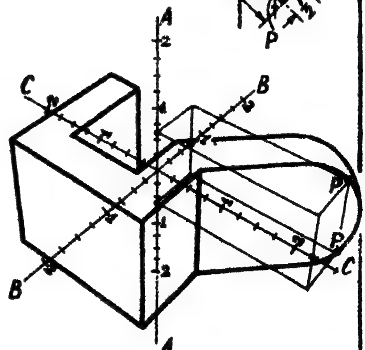



Fig. 5.

**Shade Lines** are applied to orthographic views to give them a more effective appearance. Contour lines are selected, as discussed below, and thickened to indicate that the surfaces represented by them are in shade, the general effect being to show the sides of the object very much foreshortened and to convey an impression of solidity. Shaded drawings are largely used in the technical press, in which one-view drawings predominate; their purpose is to assist the general reader to visualize readily the form of a perhaps unfamiliar object.* It is, however, the exception rather than the rule for working drawings to be shadow lined. In the exercises given herein, shadow lining may be applied to some of the more advanced work by way of experiment.

**Method.**—*The object is assumed to* be illuminated by parallel rays of light emanating from a source situated to the front and left of the object, the inclination of the rays being such that in plan and elevation their projections make 45° with *xy*. One such ray is shown in fig. 1, together with its projections. The bracket shown will have certain surfaces in shade, as indicated in fig. 1; these surfaces are represented in plan, elevation, and end view by the thickened lines in fig. 2 (the direction of the arrow in the end view being the direction of the projection of the ray after rabatment).

The treatment for the cylindrical bosses, the hole, and the recess in the base should be noted. The line *ab* is not thickened, for it represents the contour of the curved surface of a cylinder and this will not appear foreshortened: in practice, however, such lines are made a little heavier for the sake of uniformity.

Shade lines are always placed so that their thickness is additional to the thickness of the part represented, i.e. they are always placed on the outside of the outline of a drawing.

**Conventional Practice.** — The shade-line method illustrated in fig. 2 is not that adopted in modern practice. It is preferred now to take the same direction for the projected rays *in all views*, as in fig. 3. This convention makes for simplicity of treatment, and the various views, regarded singly and not collectively, lose nothing in effectiveness. As stated above, lines representing the contours of cylinders are half thickened-in, as shown by *ab* in fig. 3.

In all views, then, the right-hand and lower lines are thickened. Shade lines for circles should lessen gradually in thickness and terminate on a 45° diameter. Various examples are illustrated in the small drawings in fig. 4. These should be carefully studied with a view to accounting satisfactorily for the shade lines shown.

In all figures except fig. 5 the thickness of the shade lines has been exaggerated for purposes of demonstration.

## EXERCISES

(1) Copy in lines of uniform thickness, and to any suitable scale, the small drawings in fig. 4. Without reference to the book, insert all necessary shade lines. Compare results.

(2) Fig. 5 shows a one-view drawing, shaded in the conventional way, of the cottered connexion shown pictorially on p. 67 (fig. 3). Using fine lines throughout, copy the drawing, making the copy twice the size of the given drawing. Insert shade lines without reference to the book.

(3) Draw the views required in Examples 1, 2, and 3 (p. 10) and add shade lines.

* Refer to the excellent drawings in *Engineering* and *The Engineer*.

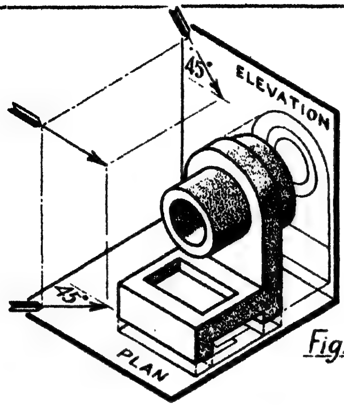



*Fig. 1.*

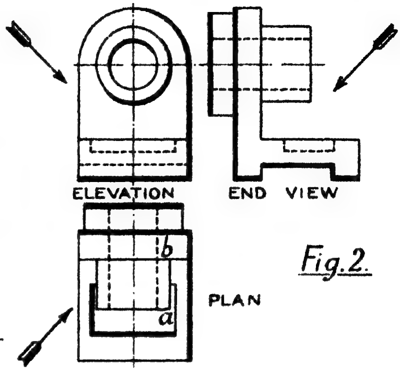



*Fig. 2.*

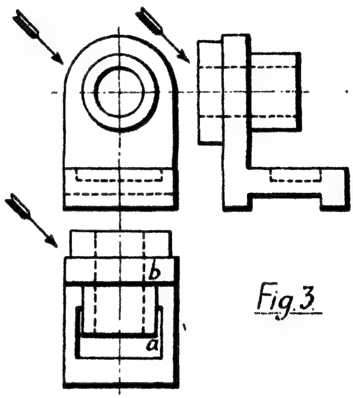



*Fig. 3.*

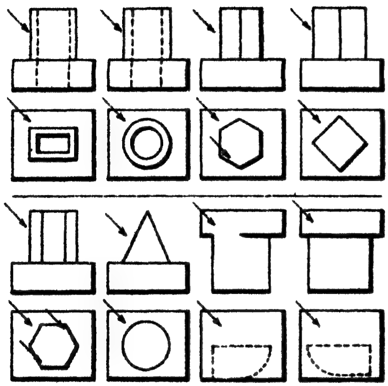


*Fig. 4.*

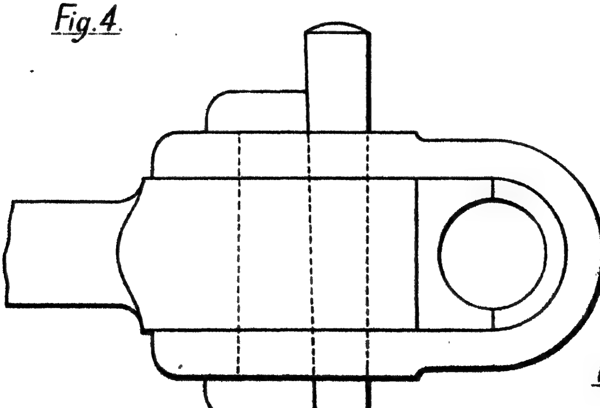


*Fig. 5.*

**A Working Drawing** is one from which an object is manufactured. It must present fully dimensioned views of the object which admit of no doubt or ambiguity, and which can be easily "read" by workmen. Among other information, the drawing should give:—the degree of finish for the various surfaces; tolerances for important dimensions; the materials to be used; details of any special operations involved. It should be correlated by means of notes with any gauges or jigs that are to be used in the process of manufacture.

The specimen drawing opposite, of an eccentric strap, illustrates the various B.S.I. recommendations for conventional practice, as set out on this page and elsewhere.

NOTE.—The drawing as shown is not very suitable for the production of the article in large numbers. The treatment for repetition work, requiring the close specification of limits and finishes, &c., is dealt with on pp. 160 to 167.

### B.S.I. Conventional Practice.

**Sizes of Drawings and Tracings.**—Standard sizes, in inches, are:—(*a*) Overall size: 72 × 40, 60 × 40, 53 × 30, 40 × 30, 40 × 27, 40 × 15, 30 × 22, 30 × 20, 27 × 20, 20 × 15, 15 × 10, 13 × 8, 10 × 8. (*b*) Minimum size between borders: 70 × 38, 58 × 38, 52 × 29, 39 × 29, 39 × 26, 39 × 14, 29 × 21, 29 × 19, 26 × 19, 19 × 14, 14½ × 9½, 12½ × 7½, 9½ × 7½. The overall sheet size should be marked on the drawing.

**Line Thickness** (*summarized*).—Refer to the various drawings herein. (*a*) Outline, thick; (*b*) dimension, extension and projection lines, thin; (*c*) centre and locus lines, thin long chain; (*d*) cross-hatching, thin; (*e*) adjacent parts, thin short chain; (*f*) hidden details, thin short dashes; (*g*) section line, thin long chain; (*h*) boundary line, thick wavy, or short zigzags joining ruled lines.

**Title Block.**—This must be placed in the lower right-hand corner. The drawing number should also be placed in the upper left-hand corner, for filing purpos The arrangement shown opposite is typ cal for general engineering drawings a should be carefully studied.

**Scale of Drawing.**—To be shown und the title. Wherever possible drawings machine parts should be full size. Thr ways of stating the scale of a drawing ar (*a*) half size, quarter size, eighth size, &c (*b*) 6" = 1 foot, 3" = 1 foot, 1½" = 1 foo &c.; (*c*) 1/2, 1/4, 1/8, &c.

**Machining Symbols.**—A surface to b machined is indicated by placing a sma triangle with a corner touching the su face (or linked to it), as shown opposit If the machining process is also to b indicated, a letter symbol should b placed within the triangle according t the following code (see symbols in right hand view). Machining all over is bes indicated by a note.

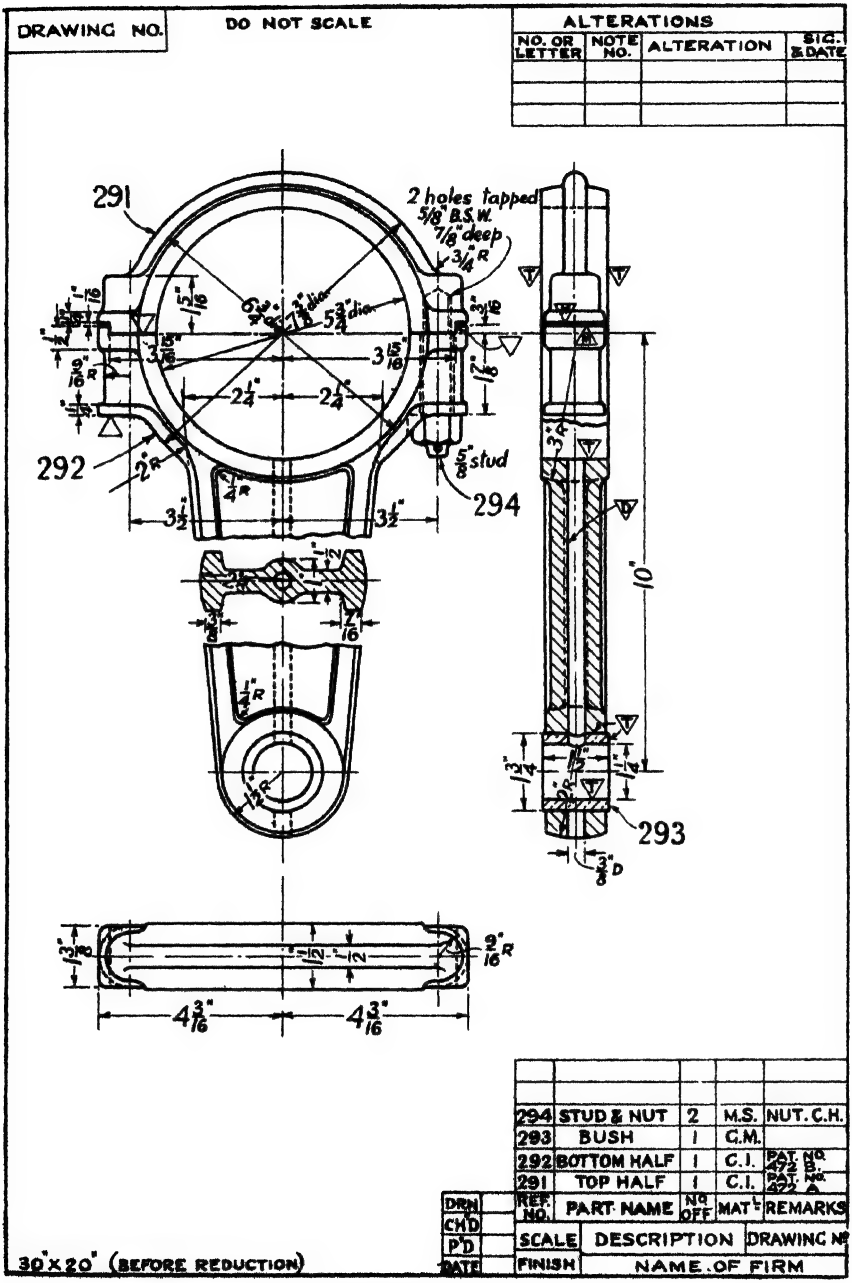
| | |
|---|---|
| B = Bore. | L P = Lap. |
| B R = Broach. | L I = Linish. |
| B F = Buff. | M = Mill. |
| B U = Burnish. | P = Plane. |
| D = Drill. | P O = Polish. |
| D B = Diamond bore. | R = Ream. |
| D T = Diamond turn. | R G = Rough grind. |
| F = File. | R M = Rough machine |
| F C = Flamecut. | S = Scrape. |
| G = Grind. | S F = Superfinish. |
| H = Hone. | T = Turn. |

(If the grade of surface finish is to b specified, numbers may be shown abov the triangle as recommended in B.S 1134.)

**Abbreviations.**—The following are a few of the abbreviations which may b used on drawings:

| *General.* | *Materials.* |
|---|---|
| British Standard, B.S. | Aluminium, Al. |
| Case Hardened, C H. | Cast Iron, C I. |
| Centre Line, C L or ℄. | Malleable Cast Iron, M C I. |
| Chamfered, chamf. | Wrought Iron, W I. |
| Circular Pitch, C P. | Cast Steel, C S. |
| Countersunk, csk. | Forged Steel, F S. |
| Diameter, dia. | Mild Steel, M S. |
| Diametral Pitch, D.P. | Nickel Steel, Ni S. |
| Not to scale, N T S. | Brass, Br. |
| Pitch Circle, P C. | Gun metal, G M. |
| Radius, rad or R. | Naval Brass, N Br. |
| Round, rd. | Phosphor Bronze, Phos B. |
| Standard, std. | White Metal, W M. |

Abbreviations are not used for steel and bronze.


DRAWING NO.
DO NOT SCALE
ALTERATIONS
NO. OR LETTER
NOTE NO.
ALTERATION
SIG. & DATE
291
292
293
294
2 holes tapped 5/8" B.S.W. 7/8" deep
5/8" stud
10"
294 STUD & NUT 2 M.S. NUT. C.H.
293 BUSH 1 G.M.
292 BOTTOM HALF 1 C.I. PAT. NO. 472 B.
291 TOP HALF 1 C.I. PAT. NO. 472 A.
REF. NO.
PART NAME
NO. OFF
MATL.
REMARKS
DRN
CH'D
P'D
DATE
SCALE
DESCRIPTION
DRAWING NO.
FINISH
NAME OF FIRM
30" x 20" (BEFORE REDUCTION)

## RIVETED JOINTS

**Rivets** are used to connect together permanently two or more plates or other pieces of metal. They are machine-forged with a single head; when used the head is held in place and a second head is formed from the projecting shank, either by hand or by machine. Wherever possible, rivets are "closed" red hot under hydraulic pressure.

**Forms of Rivet Heads for Boilers** are shown opposite. They have been standardized (B.S. 425); the proportions given are suitable for rivets from ½" to 2" in diameter, and are in terms of the nominal diameter D of the rivet. The small conical portion under the head has an angle of 60° in all cases; it serves to provide a staunch joint.

**Rivet Heads for Structures** (B.S. 275) differ in minor respects from those shown. The shank is parallel right to the head, i.e. the small 60° conical portion is omitted. Otherwise, snap, pan (type 1), and rounded countersunk heads have the proportions shown. There are in addition *flat* countersunk 60° heads; and both rounded and flat countersunk 45° heads. One type of standardized pan head has a slightly tapered end (15° only) under the head extending for a distance equal to ¼ D.

**Small Rivets.**—B.S. 641 gives standardized forms for general purpose rivets of less than ½" diameter. In addition to snap, pan, and countersunk heads there are mushroom and flat heads. In general, heads for small rivets are proportionately larger than those for large rivets.

**Lap and Butt Joints.**—In lap riveting, one plate overlaps the other and the rivets pass through both plates. In butt riveting, the plates are kept in alignment and a butt strap or cover plate is placed over the joint and riveted to each plate: frequently two straps are used, placed one on each side of the plates.

**Proportions.**—In the best class of work the rivet holes are ***drilled,*** and the operation imposes no limit on their diameter. If, however, they are ***punched,*** the diameter has a minimum value.

Let $f_c$ = crushing stress for punch, $f_s$ = shearing stress for plate, $d$ = rivet dia., and $t$ = plate thickness. Then for equilibrium:

(Area of punch) $\times f_c$ = (area of plate section sheared) $\times f_s$,

$$\text{i.e. } \tfrac{1}{4}\pi \,.\, d^2 \,.\, f_c = \pi \,.\, d \,.\, t \,.\, f_s.$$

Taking $f_c = 4f_s$, then $d = t$.

*Hence if d is less than t, the punch will fail by crushing.*

In practice, the rivet hole diameter

$$\mathbf{d = 1{\cdot}2\sqrt{t} \text{ approx.*}} \quad . \quad (1)$$

The rivet diameter chosen is usually $\frac{1}{16}$" less than that of the hole for rivets ¾" in diameter and upwards. The rivet is assumed to fill the hole when "closed", and the diameter of the hole is taken for calculations on the strength of joints.

The distance from centre to centre of two rivets *in the same row*, called the **pitch**, is determined by considerations of strength, as shown later: its minimum value = $2d$, this spacing permitting the formation of the heads. The **margin** $m$, the distance between the edge of the plate and the rivet hole, must be sufficient to prevent the plate from splitting at the edge during punching or riveting, and in practice it is made equal to $d$.†

### EXERCISES

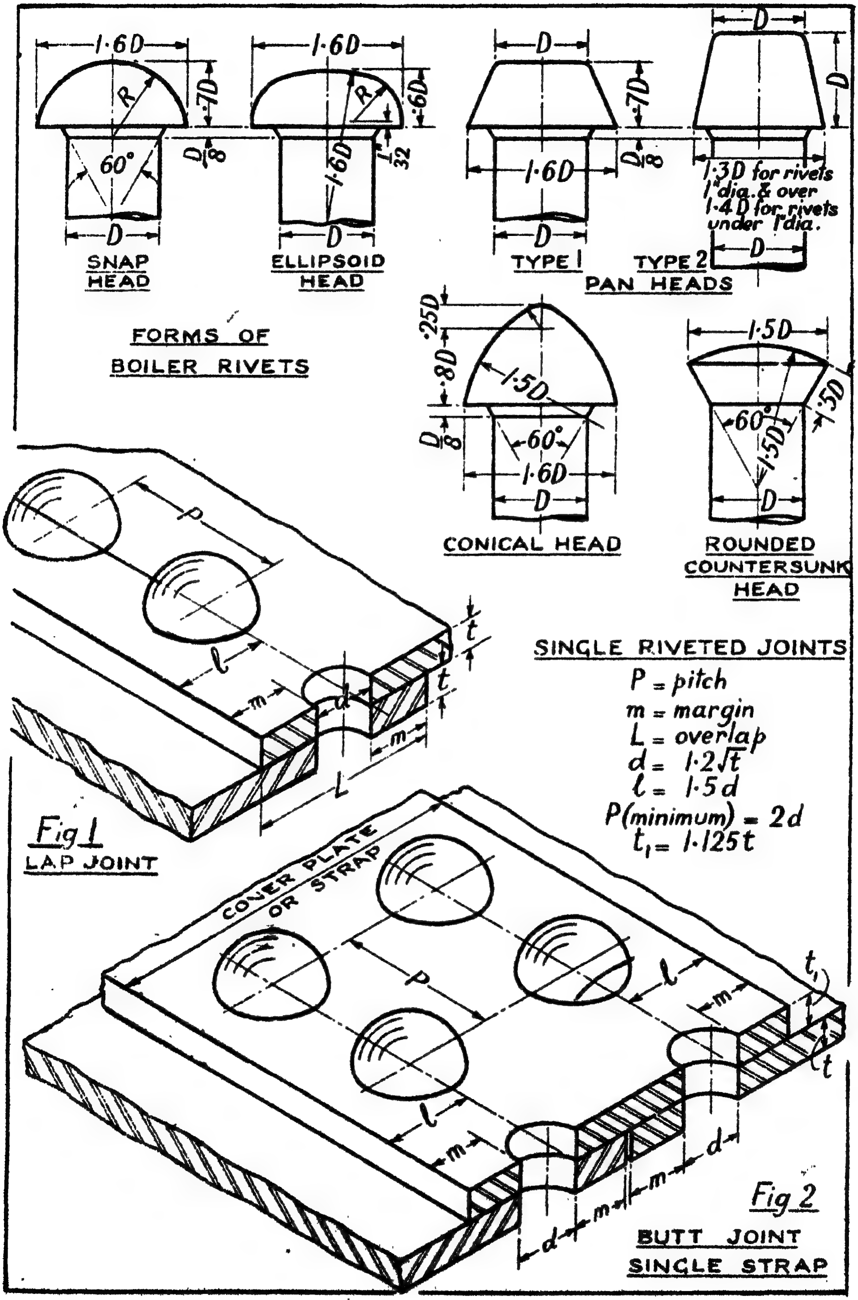
**In each exercise give a sectional elevation through the centres of the rivets, as in figs. 1 and 2, and a plan showing at least three rivets in the same row. Show the rivets in position in the section; omit the small conical neck.**

**(1) Make full-size dimensioned drawings of a single riveted lap joint suitable for two plates each ½" thick; use cup-headed rivets, pitch 1⅞".**

**(2) Make dimensioned drawings, half size, of a single riveted butt joint suitable for ¾" plates, using rivets with pan and cup heads, pitch 3". Show a single strap, 1" thick ($t_1$, fig. 2).**

* Tables of values of $d$ and $t$, such as those issued by the boiler insurance companies, are to be preferred to a formula.

† For ship plates $m = d + \frac{1}{8}$".


1·6D
·7D
R
60°
D/8
D
SNAP HEAD
1·6D
·6D
R
1·6D
1/32
D
ELLIPSOID HEAD
D
·7D
1·6D
D/8
D
TYPE 1
D
D
1·3D for rivets 1" dia. & over
1·4D for rivets under 1" dia.
D
TYPE 2
PAN HEADS
FORMS OF BOILER RIVETS
·25D
·8D
1·5D
D/8
60°
1·6D
D
CONICAL HEAD
1·5D
·5D
60°
1·5D
D
ROUNDED COUNTERSUNK HEAD
SINGLE RIVETED JOINTS
P = pitch
m = margin
L = overlap
d = 1·2√t
ℓ = 1·5d
P(minimum) = 2d
t₁ = 1·125t
p
ℓ
m
d
t
t
m
L
Fig 1
LAP JOINT
COVER PLATE OR STRAP
p
ℓ
m
t₁
t
ℓ
m
d
m
m
d
m
Fig 2
BUTT JOINT SINGLE STRAP

**Double- and Triple-riveted Plate Joints.**—The joints shown in figs. 1 and 2 on the previous page are single-riveted. Double-riveted *lap* joints have *two* rows of rivets; double-riveted *butt* joints have two rows of rivets on each side of the joint. Similarly, in triple riveting (see also page 42). three rows are provided. Quadruple- and quintuple-riveted joints are also used, generally only with butt joints.

The rivets are usually placed zig-zag, giving what is called **diagonal riveting.** In **chain riveting** the rivets are arranged directly opposite each other.

The pitch of the rivets may be settled by considering the strength of the joint, as will be discussed later. In certain classes of work, however, empirical rules are used for the spacing of the rivets: for example, in boiler work, rules drawn up by the Board of Trade or the Boiler Insurance Companies are generally adopted—(refer to fig. 3, page 41). Standardized Riveted Joints for Lancashire and Cornish Boilers are dealt with in B.S. 537.

The **distance between the rows,** $P_r$, should not be less than 0·6 × pitch for *diagonal* riveting, and 0·8 × pitch for *chain* riveting.

The plate edges for boilers, tanks, &c., are usually bevelled to an angle of 80° to facilitate **fullering** and **caulking,** operations in which the edges are driven in by a blunt tool to close the joint.

**Lap Joints.**—Examples of double- and triple-riveted lap joints are shown in figs. 1 and 2. In each, the rivet diameter and plate margin are settled by the rules given on the previous page. For boiler joints the pitch P is commonly $3d$ for double, and $3{\cdot}5d$ to $4d$ for triple riveting, where $d$ is the diameter of the rivet holes. The formation of the head requires that the least distance between two rivets, measured centre to centre, shall not be less than $2d$, and this settles the minimum value of the diagonal pitch $P_d$, fig. 1;* in practice, however, the distance between the rows, $P_r$, is made not less than $2d$.

In all lap joints the straining forces in the plates are not in the same plane and produce a couple, tending to bend the joint. The same tendency occurs in butt joints with single straps. The bending action may be avoided in butt joints by using *two* straps, as in fig. 3.

**Butt Joints.**—Types of double-riveted butt joints, each with two straps, are shown in figs. 3 and 4. The strap thickness $t_1$ varies from $0{\cdot}7t$ to $0{\cdot}8t$, when $t$ is the plate thickness. In the arrangement shown in fig. 4 the pitch of the outer rows is double that of the inner, and one butt strap (the outer strap in a boiler or tank) is narrower than the other. The advantage of this arrangement is discussed on page 40.

## EXERCISES

In each of the following draw, full size, a sectional elevation taken across the rows of rivets, as opposite, and a part plan. Obtain the diameters of the rivet holes from the formula $d = 1{\cdot}2\sqrt{t}$, increasing the dimension to the nearest $\frac{1}{16}''$. Dimension the views.

(1) Double-riveted lap joint for $\frac{1}{2}''$ plates, pitch $P = 2\frac{5}{8}''$.

(2) Triple-riveted lap joint for $\frac{5}{8}''$ plates, pitch $P = 3\frac{1}{2}''$.

(3) Double-riveted butt joint for $\frac{5}{8}''$ plates, as in fig. 4, pitch $P = 4\frac{1}{2}''$.

* A commonly accepted rule, based on Prof. Kennedy's investigations, is to make the diagonal pitch $P_d = (2P + d) \div 3$.

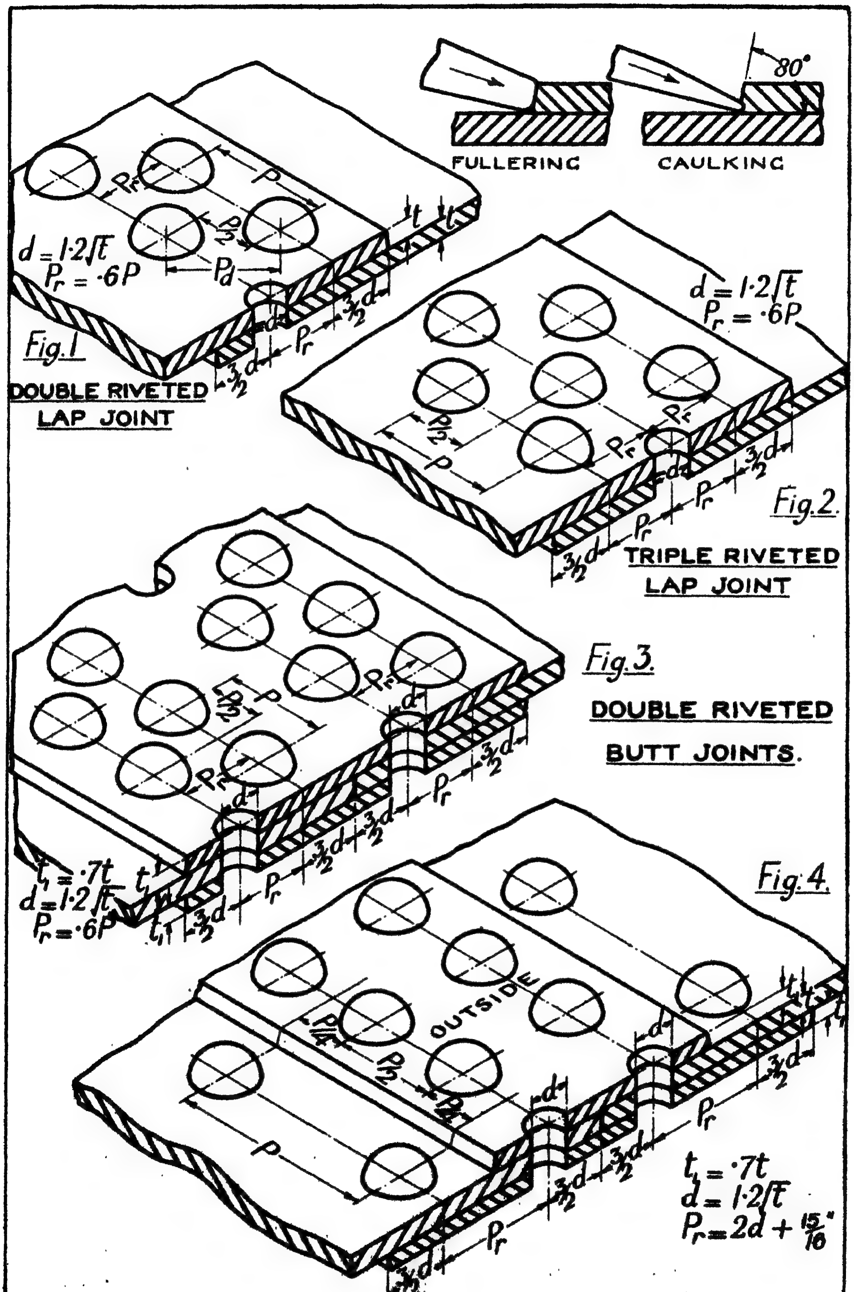



Fig. 1. DOUBLE RIVETED LAP JOINT

Fig. 2. TRIPLE RIVETED LAP JOINT

Fig. 3. Fig. 4. DOUBLE RIVETED BUTT JOINTS.

**Strength of Riveted Joints.**—When a rivet is "closed", an indeterminate stress is set up in the shank owing to its tendency to shorten on cooling: this initial stress may be considerable if the rivet is long. For this reason rivets in a joint are usually arranged so that they are subjected to shearing forces only. Rivets are in **single shear** when the plates tend to shear the rivets in one plane, figs. 1 and 7;* they are in **double shear** when the tendency is to shear them in two planes, figs. 2 and 8. The rivet area resisting shear in fig. 8 is twice that in fig. 7.

**Basis of Design.**—The resistance to breaking of the joints shown in figs. 1 and 2 may be investigated by considering a "pitch length" of each joint, figs. 3 and 4, and equating the straining force T, carried by the strip, to the product (stress in material × area resisting rupture). To each rivet there is a corresponding plate section of $(P - d)t$. Let $f_t$ = tensile stress in the plates, $f_s$ = shear stress in the rivets, and $f_c$ = crushing stress between rivet shank and plate.

***The joint may fail in the following ways:***

**(1)** Tearing of plate at rivet hole, figs. 5 and 6:—

Here $T = (P - d)t \,.\, f_t$. . . (1)

**(2)** Shearing of rivet:—

In single shear, fig. 7,

$$T = \tfrac{1}{4}\pi \,.\, d^2 \,.\, f_s. \qquad (2a)$$

† In double shear, fig. 8,

$$T = 2(\tfrac{1}{4}\pi \,.\, d^2 \,.\, f_s). \qquad (2b)$$

**(3)** Deformation under plastic flow, fig. 9:—

Experiment shows that the crushing stress, or stress at which plastic flow occurs, $f_c$, is proportional to the *projected* area of the indenting body.

Hence $T = d \,.\, t \,.\, f_c$. . . (3)

**(4)** Fracture at margin, fig. 10:—

As already stated, this does not occur if

$$m = \text{or} > d. \qquad (4)$$

Evidently, if the joint is to be equally strong in resisting tearing, shearing, and crushing, equations (1), (2*a*) or (2*b*), and (3) must be satisfied.

Hence for **Single Shear**

$$(P - d)t \,.\, f_t = \tfrac{1}{4}\pi \,.\, d^2 \,.\, f_s = d \,.\, t \,.\, f_c. \qquad (5)$$

and for **Double Shear**

$$(P - d)t \,.\, f_t = \tfrac{1}{2}\pi \,.\, d^2 \,.\, f_s = d \,.\, t \,.\, f_c. \qquad (6)$$

From (5),

$$(P - d)t \,.\, f_t = \tfrac{1}{4}\pi \,.\, d^2 \,.\, f_s,$$

so that

$$P = (\tfrac{1}{4}\pi \,.\, d^2/t \,.\, f_s/f_t) + d, \qquad (7)$$

which gives the pitch for single riveted joints in single shear.

Similarly, for joints in double shear,

$$P = (\tfrac{1}{2}\pi \,.\, d^2/t \,.\, f_s/f_t) + d. \qquad (8)$$

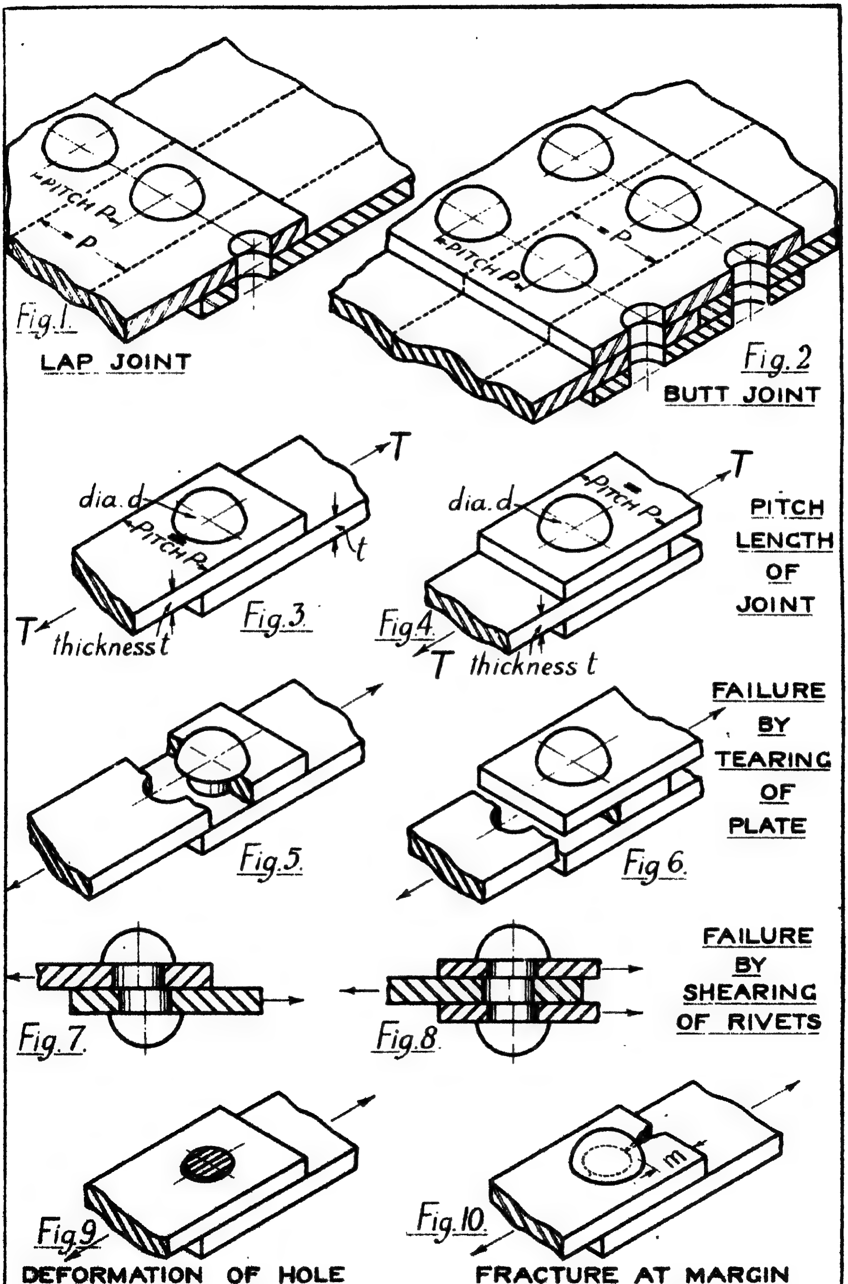
**Note.**—P must not be made less than $2d$.

**Alternative Basis of Design.**—If the rivets fill the holes when hot, it has been argued that they cannot perfectly fill them when cold; and if this assumption is valid it follows that the plates must slip slightly before the rivets are placed in shear. The slipping is resisted by friction between the plate surfaces, which are forced together when the rivets cool: any such relative motion would destroy the staunchness of a pressure joint. In an alternative method of design, developed by Bach, the joints are proportioned *to prevent slipping*—the shearing resistance of the rivets being neglected. The results, however, do not differ greatly from those arrived at by the first-mentioned method.

**Exercises.**—See page 40.

* The rivets in figs. 1 and 7 are also subjected to a small bending moment, as previously pointed out, but this is neglected in what follows.

† For steam boilers it is customary to allow a rivet in double shear to take only 1·875 the load allowed in single shear.


PITCH P
P
Fig.1.
LAP JOINT
PITCH P
P
Fig.2
BUTT JOINT
T
dia.d
PITCH P
t
T
thickness t
Fig.3.
dia.d
PITCH P
T
Fig.4.
T
thickness t
PITCH
LENGTH
OF
JOINT
Fig.5.
Fig 6.
FAILURE
BY
TEARING
OF
PLATE
Fig.7.
Fig.8.
FAILURE
BY
SHEARING
OF RIVETS
Fig.9
DEFORMATION OF HOLE
m
Fig.10.
FRACTURE AT MARGIN

**Design of Triple-riveted Butt Joint. Fig. 1.**—The determination of $t$, the plate thickness, for pressure vessels, is fully discussed on page 42. The rivet diameter $d = 1{\cdot}2\sqrt{t}$, and the thickness of the butt straps $t_1 = {\cdot}7t$ to ${\cdot}8t$. The pitch P is obtained from equations corresponding to (7) and (8) on page 38.

Consider a pitch length of the joint, fig. 2.

Resistance to tearing between rivet holes $= (P - d)t \,.\, f_t$. . . (1)

Resistance to shearing by the rivets $= 2(3 \,.\, \frac{1}{4}\pi \,.\, d^2 \,.\, f_s)$ . . (2)

(there being 3 rivets in double shear).

Equating (1) and (2),

$$(P - d)t \,.\, f_t = 2(3 \,.\, \tfrac{1}{4}\pi \,.\, d^2 \,.\, f_s)$$

and

$$P = 2(3 \,.\, \pi/4 \,.\, d^2/t \,.\, f_s/f_t) + d. \quad . \quad (3)$$

Similarly the general formulæ for pitch are:

Double shear—

$$P = 2(N \,.\, \pi/4 \,.\, d^2/t \,.\, f_s/f_t) + d,$$

Single shear—

$$P = (N \,.\, \pi/4 \,.\, d^2/t \,.\, f_s/f_t) + d,$$

where N is the number of rivets in a pitch length of one plate.

**Stress Values.**—Working values of the stresses $f_s$ and $f_t$ for steel are:—

Plates: $f_t = 12{,}000$ lb./in.².

Rivets: single shear, $f_s = 9500$ lb./in.²; double shear, $f_s = 8750$ lb./in.².

Hence $f_s/f_t = 0{\cdot}79$ for single shear and $0{\cdot}73$ for double shear. The compressive stress $f_c$ should not exceed 16,000 lb./in.² for single shear and 21,000 lb./in.² for double shear. For the joint considered, its value is given by the equation:

$(P - d)t \,.\, f_t = 3 \,.\, d \,.\, t \,.\, f_c$ . . . (compare with (5) on previous page). . (4)

**Efficiency of the Joint** may be defined as the ratio, strength of weakest section in a pitch length of joint to the strength of the same length of solid plate. The resistance to fracture of the solid plate is $P \,.\, t \,.\, f_t$, and if equation (3) is satisfied, then

**Efficiency** $= (P - d)t \,.\, f_t \div P \,.\, t \,.\, f_t$
$= (P - d) \div P$. . (5)

If the pitch is made smaller than that given by (3), the same expression (5) holds for efficiency. If, however, the pitch is increased (and this is rare) the joint will fail by shearing the rivets; then, the efficiency is the ratio, resistance to shear of rivets to the resistance to tearing of the solid plate, i.e.

Efficiency $= 2(3 \,.\, \pi/4 \,.\, d^2 \,.\, f_s)$
$\div P \,.\, t \,.\, f_t$. . (6)

**Alternative Design.**—A stronger joint is given by omitting alternate rivets from the outer row, as in fig. 3. For a pitch length of the joint:—

Resistance to tearing between rivet holes $= (P - d)t \,.\, f_t$, as in (1). (7)

Resistance to shearing by rivets $= 2(5 \,.\, \frac{1}{4} \,.\, \pi \,.\, d^2 \,.\, f_s)$ . . . . (8)

Equating (7) and (8),

$$P = 2(5 \,.\, \pi/4 \,.\, d^2/t \,.\, f_s/f_t) + d. \quad (9)$$

The pitch given by this formula is usually reduced for pressure vessels to give staunchness.

This type of joint might fail by the plate tearing between an inner row and by shearing the outer row of rivets: the resistance to failure in this way is $\{(P - 2d)t \,.\, f_t + 2 \,.\, \frac{1}{4} \,.\, \pi \,.\, d^2 \,.\, f_s\}$. If equation (9) is satisfied the efficiency may be either $(P - d) \div P$, or $\{(P - 2d)t \,.\, f_t + 2 \,.\, \frac{1}{4} \,.\, \pi \,.\, d^2 \,.\, f_s\} \div P \,.\, t \,.\, f_t$, *whichever is the smaller.*

It is usual to make the distance $P_r$ between the inner rows less than $P_a$ the distance between the outer rows, and Board of Trade Rules for these distances, also for the thickness of the butt straps, are given opposite.*

## EXERCISES

Design the following joints and prepare dimensioned drawings of them, showing cup-headed rivets. Calculate the efficiency of each joint.

(1) Triple-riveted butt joint, as in fig. 1, for plates 1″ thick.

(2) Double-riveted butt joint, as in fig. 3, p. 37, for plates $\frac{3}{4}$″ thick.

(3) Triple-riveted butt joint, as in fig. 3, for plates $\frac{11}{16}$″ thick. Calculate the pitch but reduce it to 8″ for staunchness.

*Answers.*—(1) $t_1 = \frac{3}{4}$″, $d = 1\frac{1}{4}$″, $P = 6\frac{1}{2}$″, *Eff.* = 81 *per cent*; (2) $t_1 = \frac{9}{16}$″, $d = 1\frac{1}{16}$″, $P = 4\frac{1}{2}$″, *Eff.* = 76·4 *per cent*; (3) ($P = 9{\cdot}34$″), $d = 1$″, $t_1 = \frac{1}{2}$″, $P_r = 2$″, $P_a = 2\frac{3}{4}$″, *Eff.* = 87·5 *per cent.*

* Refer to Examination Question No. 3, p. 216.

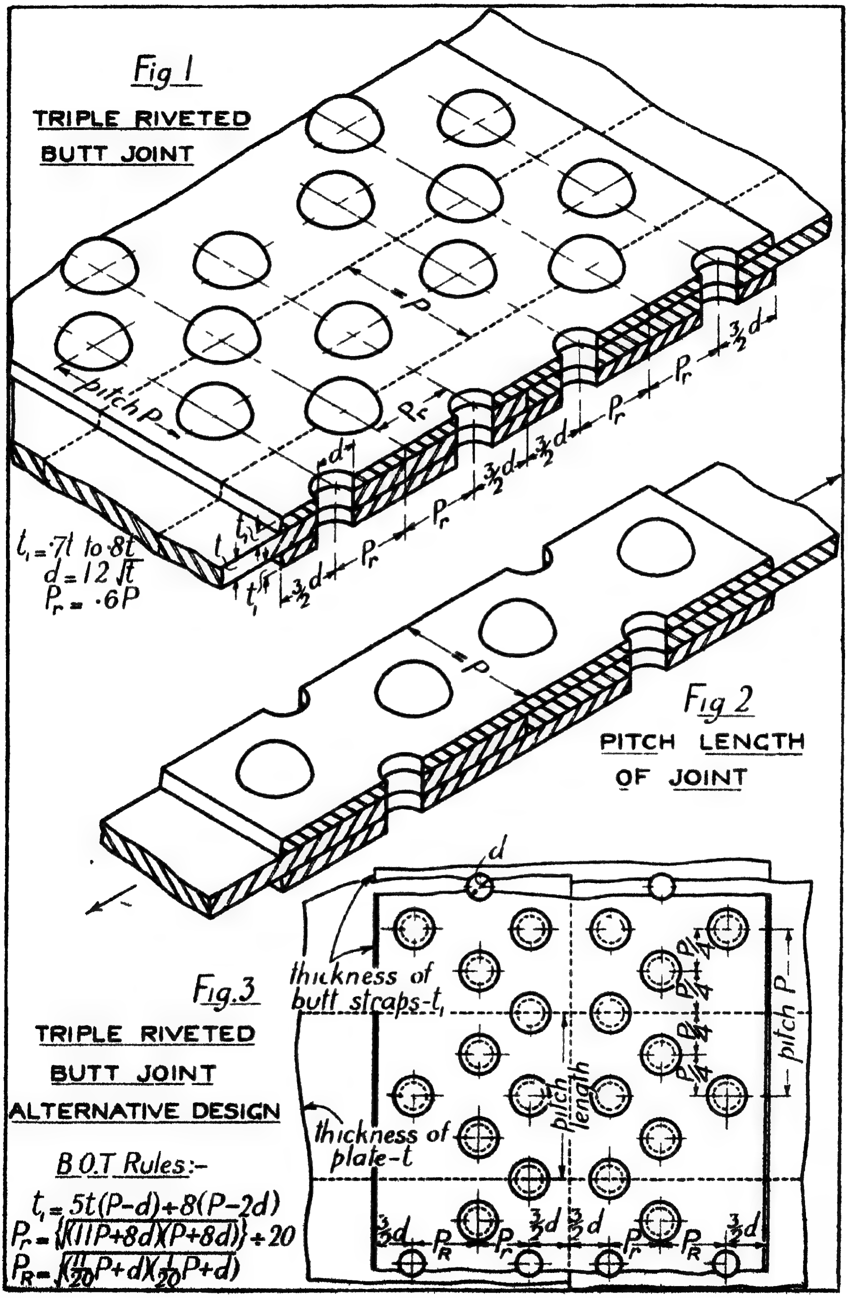


Fig 1
TRIPLE RIVETED
BUTT JOINT
$t_1 = \cdot 7t$ to $\cdot 8t$
$d = 1{\cdot}2\sqrt{t}$
$P_r = \cdot 6P$
pitch P
P
$P_r$
d
$\frac{3}{2}d$
Fig 2
PITCH LENGTH
OF JOINT
P
Fig.3
TRIPLE RIVETED
BUTT JOINT
ALTERNATIVE DESIGN
B.O.T. Rules:-
$t_1 = 5t(P-d) \div 8(P-2d)$
$P_r = \sqrt{(11P+8d)(P+8d)} \div 20$
$P_R = \sqrt{(\frac{11}{20}P+d)(\frac{1}{20}P+d)}$
thickness of butt straps-$t_1$
thickness of plate-t
pitch length
pitch P
$\frac{P}{4}$
$P_R$
$P_r$
$\frac{3}{2}d$

**Thin Cylinders with Riveted Joints, subjected to internal fluid pressure.** Let $p$ = pressure in lb./in.$^2$; let D = diameter, L = length, $t$ = thickness, all in inches; let $f_t$ = stress in plate in lb./in.$^2$.

**Longitudinal Section. Figs. 1 and 2.**—The bursting force on this section is the resultant of the normal pressure over one-half of the cylinder, and equals $(p \,.\, L \,.\, D)$ lb. The resistance to bursting is the product (area of metal × stress), and equals $2(t \,.\, L \,.\, f_t)$ lb. Hence, for equilibrium,

$$p \,.\, L \,.\, D = 2(t \,.\, L \,.\, f_t),$$

$$\text{i.e.} \quad p \,.\, D = 2t \,.\, f_t. \quad . \quad . \quad . \quad (1)$$

From which

$$f_t = p \,.\, D \div 2t. \quad . \quad (2)$$

If, however, the shell is riveted along this section, the plate area is not $2t \,.\, L$ but $2t \,.\, L(P - d) \div P$, where $d$ is the dia. of the rivet hole and P is the pitch. In other words, the plate area $= 2 \,.\, t \,.\, L \times$ plate efficiency of joint. Hence for a riveted longitudinal joint equation (1) becomes

$$p \,.\, D = 2 \,.\, t \,.\, f_t \times \textit{efficiency}. \quad (3)$$

**Circumferential or Ring Section. Fig. 1.**—Here the axial bursting force is the resultant of the pressure on the end, and equals $\frac{1}{4}\pi \,.\, D^2 \,.\, p$. The resistance to rupture is the product (annular area of shell × stress) $= \pi \,.\, D \,.\, t \,.\, f_t$. Hence

$$\tfrac{1}{4}\pi \,.\, D^2 \,.\, p = \pi \,.\, D \,.\, t \,.\, f_t,$$

$$\text{i.e.} \quad p \,.\, D = 4 \,.\, t \,.\, f_t. \quad . \quad (4)$$

From which

$$f_t = p \,.\, D \div 4t \text{ (compare with (2))} \quad (5)$$

The ring section is thus twice as strong as the longitudinal section.

**Joints for Cylindrical Pressure Vessels: Boilers,* Tanks, &c. Fig. 3.**

**Longitudinal Joint.**—The type of joint (i.e. lap or butt, single- or multiple-riveted) has first to be settled. It depends largely on the diameter of the vessel and the working pressure, and definite rules cannot be given. A joint efficiency must then be assumed, based upon the following values:

Lap Joints: Single 55%, double 70%, triple 77%.

Butt Joints: Single 65%, double 80%, triple 85%, quadruple 90%.

The plate thickness $t$ may be calculated from (3), taking $f_t$ = 12,000 (for steel), and the rivet diameter, pitch, &c., may then be determined as on previous pages.

**Ring Joint.**—This may be made weaker than the longitudinal joint—compare (5) with (2). It is usually of the lap type with rivets equal in diameter to those for the longitudinal joint.

The pitch may be settled for staunchness and the rivet shearing area checked. Thus, if P is the "ring pitch", and if there are $n$ rows, the total number of rivets $N = n \,.\, \pi \,.\, D \div P$. The total shear area $= N(\frac{1}{4}\pi \,.\, d^2)$, and the total resistance to shear $= N(\frac{1}{4}\pi \,.\, d^2)f_s$. The value of $f_s$ given by equating this resistance to the axial bursting force $\frac{1}{4}\pi \,.\, D^2 \,.\, p$ should not exceed 9500 lb./in.$^2$.

**Overlapping Joints** occur where longitudinal and ring joints meet, as in fig. 3. For lap joints the junction is effected as in fig. 4; for butt joints as in fig. 5. Both drawings are self-explanatory and should be carefully studied.

## EXERCISES

(1) A steel cylinder 4′ 0″ dia. has a longitudinal double-riveted butt joint, as in fig. 3, p. 37. The working pressure is 160 lb./in.$^2$. Obtain a suitable plate thickness and design the joint.

(*Answer.*—$t = \frac{7}{16}''$, $t_1 = \frac{3}{8}''$, $d = \frac{15}{16}''$, $P = 4\frac{1}{2}''$, *Eff.* = 81 *per cent.*)

(2) The cylinder in (1) has also a double-riveted lap circumferential joint. Obtain a suitable rivet pitch and make a drawing, as in fig. 4, of the junction of the three plates.

(3) A Lancashire boiler, working pressure 160 lb./in.$^2$, internal diameter 8′ 1½″, has triple-riveted butt longitudinal joints and double-riveted lap circumferential joints. Obtain the plate thickness and design the joints.

* The proportions of steam-boiler joints are usually based on empirical formulæ compiled by authorities such as the Board of Trade, Lloyds, and the boiler insurance companies. Standardised riveted joints for Lancashire and Cornish boilers are given in B.S. 537.

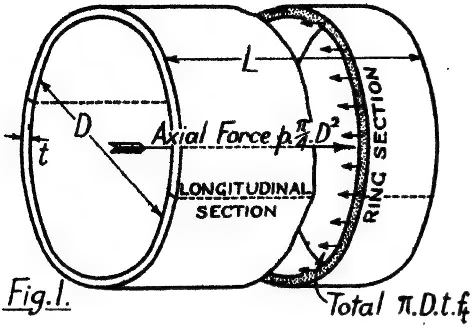



Fig. 1.

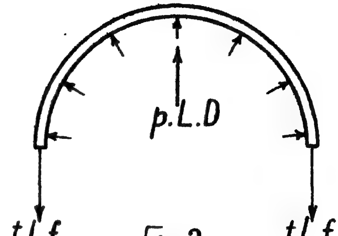



Fig. 2.

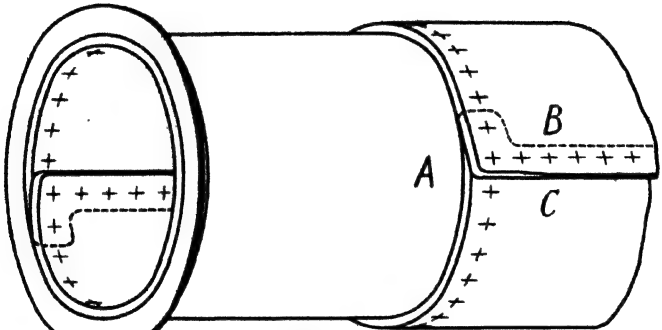



Fig. 3. OVERLAPPING JOINTS

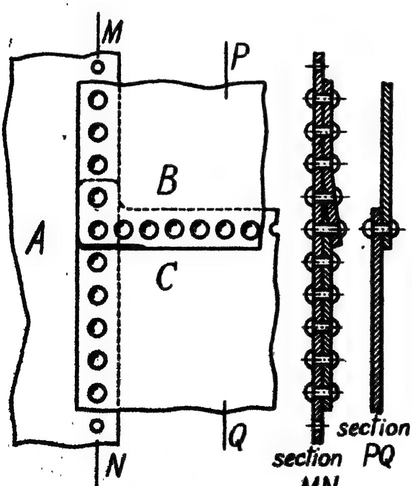



Fig. 4. LAP JOINTS

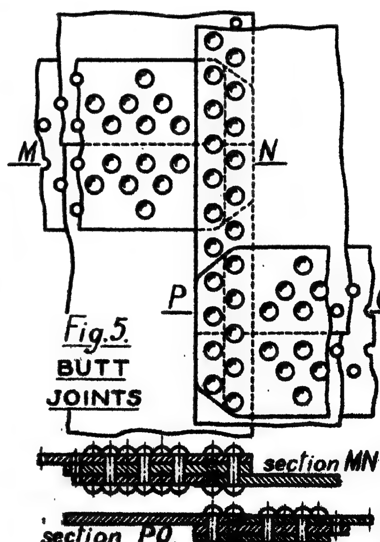



Fig. 5. BUTT JOINTS

**A Screw Thread** is formed by cutting or rolling a continuous helical groove on a cylindrical surface. The threaded portion, or screw, engages with a corresponding threaded hole in a nut or other machine part; the two elements, screw and nut, form what is called a screw pair. The thread may be right- or left-handed—figs. 1 and 2. Multiple threads, fig. 3, are also used: these are discussed on page 46.

**The Pitch** of a screw thread is defined as the distance measured along a line parallel to the axis of the screw between corresponding points on adjacent thread forms in the same axial plane. The axial advance per revolution in multiple-threaded screws is called the *lead*.

**Forms of Screw Threads.**—To permit interchangeability between nuts and screws of the same nominal dimensions, screw threads have been standardized, and the following are the principal systems in use.

**British Standard Whitworth Screw Threads (B.S.Whit.). Fig. 4.**—This system was introduced by Sir Joseph Whitworth in 1841. The angle between the slopes of the vees, measured in an axial plane, is 55°, and one-sixth of the depth of the full vee is rounded off at top and bottom. The pitch adopted is based on experience, and is suitable for both brittle and ductile materials. Details of B.S.Whit. threads for screws from ¼" to 6" diameter are given in Table 1 on page 223.

In practice the basic sizes of screw threads have to be adjusted to give clearance for assembly. British Standards Publication No. 84 includes tables of dimensions for threads for bolts and nuts to give three classes of fit: Close, Medium, and Free. For the purpose of these " tolerance zones ", the crests of internal threads are slightly flattened.

**British Standard Fine Screw Threads (B.S. Fine).**—The pitches of the Whitworth series are not fine enough for many purposes, and a system of fine pitches, of the Whitworth form, has been adopted for screws from ¼" to 4½" diameter inclusive. Details are given in Table 2, page 223.

**American National Screw Thread. Fig. 5.***—This thread has an angle of 60° and in the basic form is truncated at the root and crest by one-eighth of the full theoretical depth. In practice this theoretical depth H, for the screw, is reduced by ⅛H at the crest and by ¼H at the root, as shown at the right in fig. 5; similarly, for the mating nut, H is reduced by ¼H at the crest and by ⅛H at the root. Hence the maximum depth of engagement is ⅝H.

American National Threads are divided into two series, Coarse (N.C.) and Fine (N.F.). Full dimensions are given in the National Bureau of Standards Handbook H. 28. Design information, for Coarse Threads, is given in Table 4, page 224.

**Metric Thread (S.I.).**—The Système Internationale (S.I.) or Metric Thread is similar to the American Standard; it has an angle of 60° and is truncated. Details are given in B.S. 1095.

**British Association Screw Threads (B.A.).**—This system is adopted for small screws—below ¼" diameter. The angle of thread is 47·5° and metric dimensions are used. The theoretical depth is reduced by ·268 × pitch at both the crest and the root, giving an effective thread depth of ·6 × pitch (approx.); the crests are rounded to a radius of ·18 × pitch. Details of B.A. Threads are given in Table 3, page 224.

**Forms of Screws for Transmitting Power.**—Two important threads are shown in figs. 6 and 7: the Square Thread and the Acme Thread. With the square thread there is no oblique pressure tending to burst the nut, and the bearing surface is large. The Acme thread is easier to cut and is stronger at the root; it is sometimes used for leading screws of lathes, the half nuts engaging with it easily. In practice, the depth D is made a little greater than ½ × pitch to give clearance, so that the width R at the root is a little less than the width C at the crest. Full particulars are given in B.S. 1104.

**Exercises.**—See page 46.

* The proportions were adopted in 1933 by the American National Screw Thread Commission and supersede the earlier Standard devised by the Franklin Institute in 1864.

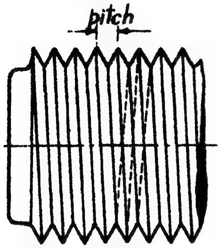



Fig 1 Right hand

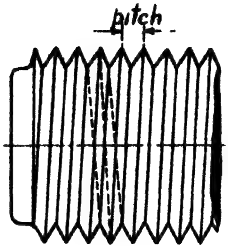



Fig 2 Left hand

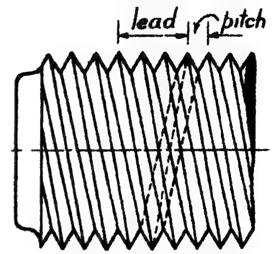



Fig 3 Right hand triple

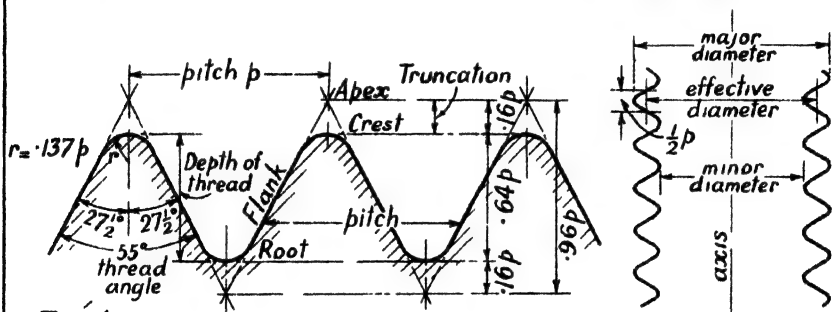



Fig 4 WHITWORTH THREAD

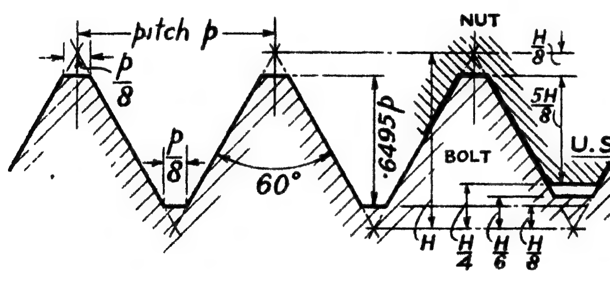



Fig. 5 U.S. STANDARD THREAD (AMERICAN NATIONAL)

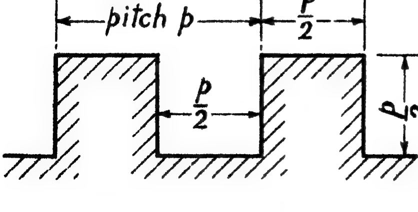



Fig. 6 SQUARE THREAD

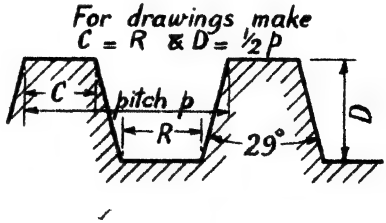



Fig. 7 ACME THREAD

**Conventional Representation of Screw Threads.**—In drawing screw threads, the true helical thread forms are rarely set out accurately, it being sufficient to show the threads by means of straight lines. The screwed bar in fig. 1, A, called a stud, is shown with vee threads joined by straight lines accurately spaced. This kind of drawing takes long to produce and it can be justified only in special circumstances, e.g. where a drawing has to be interpreted by non-technical readers or where the threads have a special form. In engineering working drawings no attempt is made to show ordinary threads at all, and fig. 1, B, shows the treatment, for the same stud, recommended by the B.S.I.: the threaded portion is indicated simply by a thin continuous line. The treatment for a screwed pipe is given in fig. 2. This is a conventional practice deliberately adopted to save time.

The stud in fig. 1, B, is repeated in fig. 3 to indicate the treatment necessary when it has to be shown in position on a composite drawing. The shorter part screwed to a depth *a* enters a hole screwed to a depth *b* (*b* being a little greater than *a* to give clearance for the stud); the hole has been drilled to a depth *c* (*c* being a little greater than *b* to give clearance for the tap).

The conventional treatment of internal threads follows the same practice, i.e. a full line along the crests of the threads and a thin line along their roots.

A compromise between the full treatment in fig. 1, A, and the extreme convention in fig. 1, B, is shown in fig. 4. The threads are indicated by straight lines, alternately thin and thick, spaced by eye and sloping slightly to indicate the "hand" of the screw. Care must be taken not to exaggerate the slope. This method of showing threads has been largely used throughout this book.

The B.S.I. recommend the following conventions for notes on drawings of screws:—

For indicating left-hand threads the abbreviation L.H. is to be used. For indicating right-hand threads the abbreviation R.H. is *not* necessary. Hence, 1" B.S.Fine means a British Standard Fine Thread, 1" dia., right-handed; similarly, 1" B.S. Whit.L.H. means a British Standard Whitworth Thread, 1" dia., left-handed.

**Screws of large pitch** may be shown fully, as in fig. 5. The thread flanks may be sloped at 60° to facilitate drawing; and, of course, the threads will be shown by straight lines. The screws in fig. 5 are *left-handed*. A little consideration will show that the internal threads in the sectioned hole must be shown sloping in the opposite direction to the external threads on the screw.

**Square Threads** may be drawn as in figs. 7 and 6, these showing respectively a right-handed square-threaded screw and, in section, the corresponding screwed hole. The construction should be evident from the figures. For small screws the construction may be still further simplified, as shown in fig. 8.

**Multiple-threaded Screws.**—In a single-threaded screw, the greater the pitch the smaller is the diameter at the bottom of the thread—and the weaker becomes the screw. When large pitches are required, therefore, it is usual to provide two or more threads of the same pitch arranged parallel to each other. A double-threaded screw is shown in fig. 9; a triple-threaded screw is shown in fig. 3, page 45. It should be noted that for an even number of threads the crests on both sides are directly opposite each other; for an odd number of threads the crest on one side of the screw is opposite the root on the other.

## EXERCISES

(1) Draw, as in fig. 5, a 4" length of a Whitworth screw, 2½" dia. Refer to Table 1, p. 223.

(2) Draw, as in fig. 7, a 5" length of a square-threaded screw, pitch 1", dia. 3". Show a screwed hole in section, as in fig. 6, passing through material 2½" thick.

(3) A square-threaded screw, 4" dia., has a double thread, pitch 1", lead 2". Draw a 6" length of the screw.

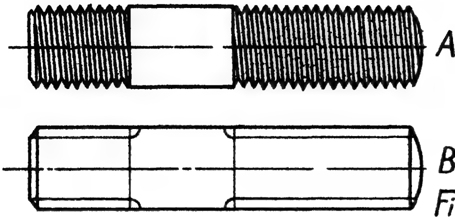



Fig 1

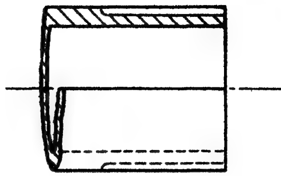


Fig 2

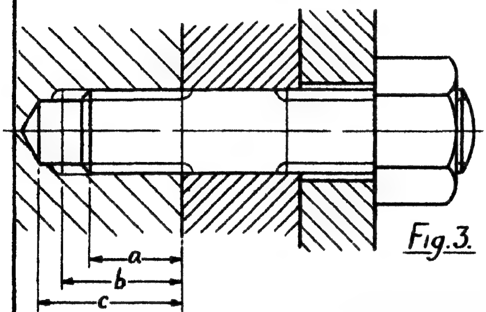



Fig.3.

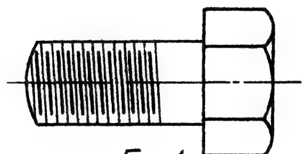
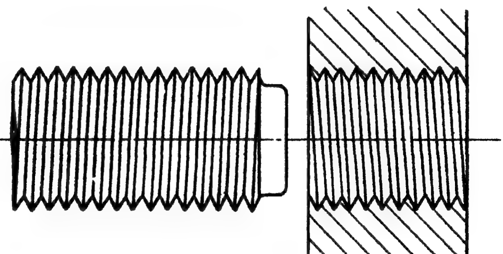


Fig.4.

Fig.5.



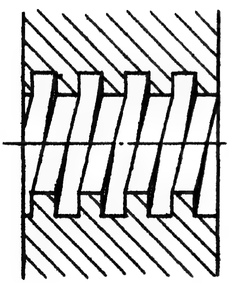


Fig.6.

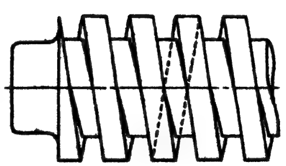


Fig.7.

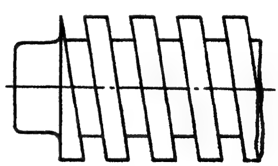
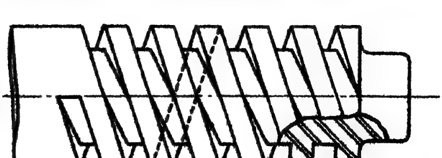


Fig.8.

SQUARE THREADS — SINGLE

Fig.9.



SQUARE THREAD DOUBLE

**Nuts and Bolt Heads** are usually made in the form of hexagonal or square prisms with the corners of one end rounded off, as shown in figs. 1 and 3. For bolts of the same diameter, the width W across the flats is the same for square as for hexagonal nuts.

Proportions of nuts and bolt heads have been standardized, and Table 6 on page 225, from B.S. 190 and B.S. 1083, gives the principal dimensions for bolt diameters from $\frac{1}{4}''$ to $2''$. The dimensions cannot be expressed exactly in terms of the bolt diameter, and *whenever clearance space is limited the true dimensions of the nut or bolt head should be taken from the table.*

The original Whitworth proportions for nuts and bolt heads are given by using Col. A in Table 6. Experience has shown that these sizes can be safely reduced, and the proportions given by using Col. B are now widely adopted. It will be noted that the reduced sizes are given by taking those for the next standard size in the Whitworth list. Whitworth proportions are, however, still in use.

Thickness of nuts and bolt heads are as under:

Whitworth (B.S. 190):
thickness of nut = dia. $d$ of bolt.
thickness of head = $\frac{7}{8}d$.

Machine Bolts (B.S. 1083):
thickness of nut = $\frac{7}{8}d$.
thickness of head = $\frac{3}{4}d$,

**Conventional Representation.**—The simplified proportions given in figs. 4 and 5 may be used for drawings where the representation of the exact size of nuts and bolt heads is immaterial. It will be seen that the width across the corners, taken as $2d$, is larger than the actual dimensions given in Table 6 (for a $1''$ bolt, the sizes are $1{\cdot}93''$ in one case and $1{\cdot}71''$ in the other, compared with $2''$ in the drawing).

The corners of the head and nut are chamfered at an angle of 30° to the end. The curves of intersection between the flat sides and the conical chamfer surface are hyperbolas: it is sufficient, however, to show the curves as circular arcs, suitable radii for which are given in the figure. If the exact sizes for nuts and bolt heads are taken from the table, the proportions for the various radii in figs. 4 and 5 may still be used. The differences between the views of the hexagonal head and nut should be carefully noted.

Conventional proportions for a square nut are shown in fig. 5. The cup or round head, fig. 2, may be given the same proportions as the round-headed screw on page 53.

**Tap Bolts and Studs.**—Bolts are widely used to fasten together machine parts, as in fig. 6. If, however, the presence of a nut or bolt head on one side is inadmissible, a tap bolt may be used, as in fig. 7; the bolt passes clear through one part and is screwed into a tapped hole in the other. An undesirable feature of the tap bolt fastening is the tendency to break the threads in the holes when the bolts are frequently removed and replaced—especially when the holes are in cast iron. This disadvantage is overcome by the use of studs, fig. 8; the stud is screwed tightly into one part by means of a stud-box (a deep nut with a screw hole passing partly through it) and is not withdrawn when the nut is removed.

## EXERCISES

***All to be drawn full size and dimensioned***

(1) Draw three views of a hexagonal bolt and nut, dia. $1\frac{1}{4}''$, length of shank $5''$, length of screw $2\frac{1}{2}''$. Show the nut half-way along the thread.

(2) Draw three views of a cup-headed bolt, dia. $1''$, length of shank $5''$, length of screw $2\frac{1}{4}''$; show a square nut on the bolt.

(3) A stud $1\frac{1}{4}''$ dia. passes through a flange $1\frac{3}{8}''$ thick and is screwed into a tapped hole $1\frac{1}{2}''$ deep, as in fig. 8. The flange is secured by a hexagonal nut bearing on a washer, and the stud is drilled for a split pin $\frac{1}{4}''$ dia. Show the arrangement in section.

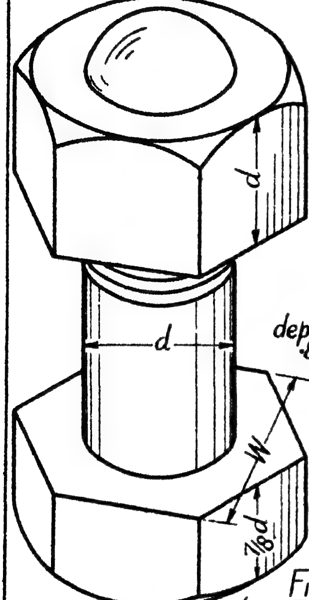



Fig. 1

HEXAGONAL BOLT HEAD & NUT

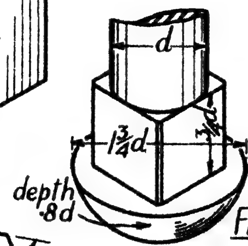



Fig. 2

CUP-HEADED BOLT WITH SQUARE NECK

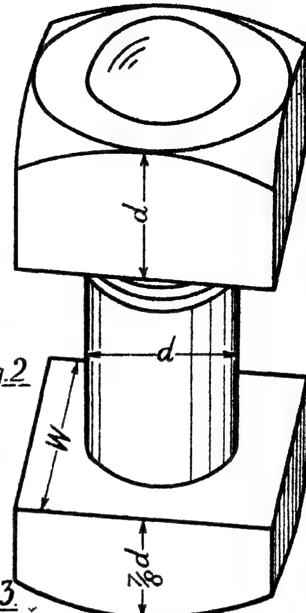



Fig. 3.

SQUARE BOLT HEAD & NUT

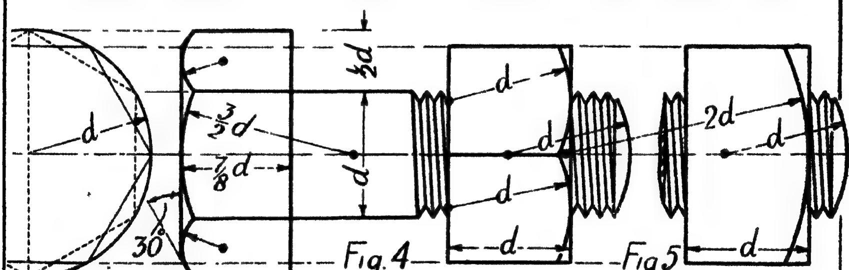



Fig. 4 HEXAGONAL — HEXAGONAL — Fig. 5 SQUARE.

CONVENTIONAL REPRESENTATION OF NUTS & BOLT HEADS.

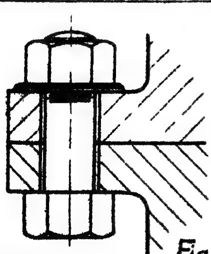


Fig. 6. BOLT & NUT

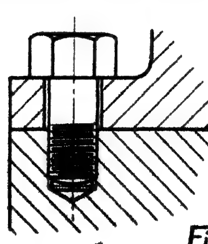


Fig. 7. TAP BOLT

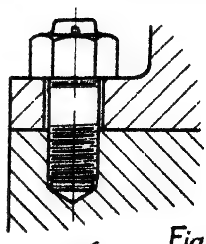


Fig. 8. STUD

There is a tendency for nuts to work loose when the bolts to which they are attached carry varying axial loads and at the same time are subjected to vibration—e.g. the bolts of an engine connecting rod. The consequences of any slackening back of the nuts may be serious, and to guard against this, various locking arrangements are used. Some of the more common of these devices will be described here; others are illustrated in the drawings elsewhere.

**Locked Nuts.**—In this method two nuts are used. One is screwed down tightly in the ordinary way. The threads of this nut will make contact with the bolt threads as shown in fig. 2, in which the thread clearance has been exaggerated. This nut is then held while the second is screwed down upon it and tightened *almost* to its limit—two spanners being used. The upper nut is now held while the lower is turned back as far as possible: evidently it can turn through a very small angle only. The threads will now be in contact as shown in fig. 3; the nuts are wedged tightly on the screw and the axial load on the bolt is carried *by the outer nut only.** For economy, the inner nut is reduced in thickness to two-thirds that of the standard nut.

The arrangement necessitates the use of a thin spanner, and as this is not always available, the thin nut is often placed on the outside. A compromise is shown in fig. 4, in which each nut is made three-fourths the thickness of an ordinary nut.

The provision of a split pin is an additional safeguard: to be effective the pin should bear on the nut.

**Castle Nut. Fig. 5.**—This arrangement is very satisfactory and is widely used for motor-car and locomotive work. A number of slots, generally six, are cut in a cylindrical part provided above the hexagonal nut, and the bolt is drilled to take a split pin. The pin can be inserted through this hole and through opposite slots at every 60° of rotation of the nut. The dimensions of castle nuts have been standardized and approximate proportions are given in the figure. Refer also to B.S. 1083.

**Slotted Nuts** are very commonly used. These are nuts of standard proportions with six slots across the upper face, as arranged for castle nuts.

**Spring Washer. Fig. 6.**—In this device a spring-steel washer is split and sprung open. When used under a nut the elasticity of the flattened spring keeps the nut tight on the bolt when the tension in the latter is diminished from any cause.

**Ring or Collar Nut.**—For large nuts, such as those used for marine connecting rods, the arrangement shown in fig. 7 is frequently adopted. The lower portion of the nut is turned and grooved, and fits into a recess in the cap. A set screw passing through the cap bears on the bottom of the groove and prevents rotation. An alternative method is shown in fig. 8, in which a ring, pinned in position, is used to take the set screw. Average proportions are given in the figure.

## EXERCISES

*Both to be drawn [illegible]*

[illegible] an elevation, half [illegible] section, an end [illegible] a pl[illegible] castle [illegible] complete wit[illegible] pin [illegible]crew 2[illegible]

[illegible] sectional elev[illegible] and a [illegible] in fig. [illegible] bolt [illegible] dia.

* In practice, how[illegible] method descr[illegible] tion is mainly of theore[illegible]

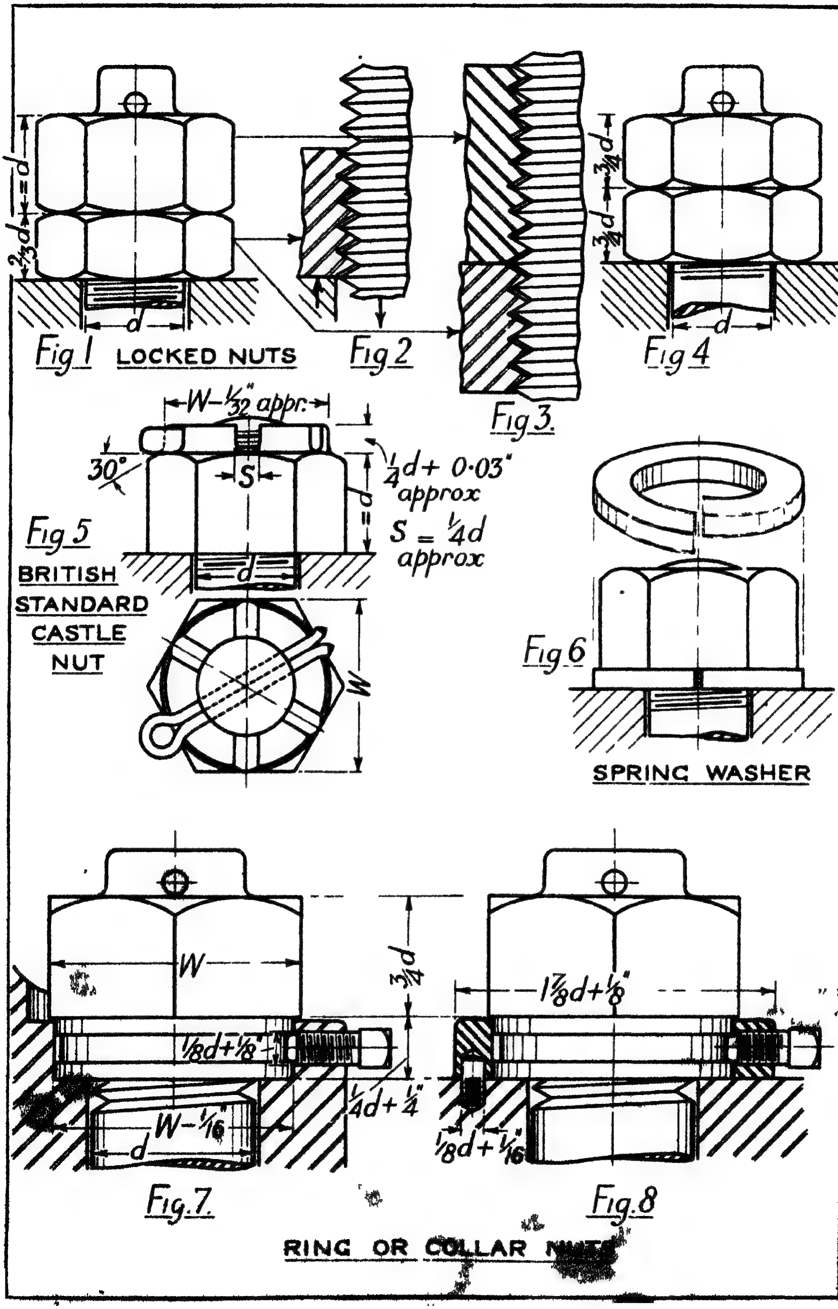


Fig 1
LOCKED NUTS
Fig 2
Fig 3.
Fig 4
d
W-1/32" appr.
30°
S
1/4 d + 0·03" approx
S = 1/4 d approx
Fig 5
BRITISH STANDARD CASTLE NUT
W
Fig 6
SPRING WASHER
W
3/4 d
1/8 d + 1/8"
1/4 d + 1/4"
W-1/16"
1 7/8 d + 1/8"
1/8 d + 1/16"
Fig.7.
Fig.8
RING OR COLLAR

**Screw Heads.**—The forms and proportions of screw heads in common use are shown in figs. 1, 2, and 3. These proportions represent the standard in Britain for B.S.Whit. and B.S.Fine screws from $\frac{1}{8}''$ to 1″ in diameter. The dimensions of the slots in the heads have been omitted: in making drawings the student may use his own judgment as to their sizes. The thread for the countersunk-headed screw is taken to the head; for the other screws, it stops at a distance from the head not exceeding twice the pitch of the thread. Standard sizes are given in B.S. No. 450.

Heads for B.A. screws are similar to those shown, although some dimensions differ slightly. In addition to the countersunk, round, and cheese heads, other forms are used: two of these, the filister head and the instrument head, are shown in figs. 4 and 5. Detailed dimensions of the heads of these and other screws are given in B.S. No. 57.

**Set Screws,** fig. 6, are used chiefly to prevent relative movement between two machine parts. They are invariably of steel, and in the best practice are case-hardened. In the **standard** form the heads are square and the points flat, both being chamfered. There are three standard sizes of square heads: small, as illustrated opposite, medium, and large. For a screw $\frac{1}{2}''$ diameter the widths across the flats of the three types are respectively 0·5″, 0·563″, and 0·75″. For details refer to B.S. No. 451. **Headless** set screws, or **Grub** screws, are used on moving parts where a projecting head would be dangerous. Two types of grub screw are shown in fig. 7, one with a conical head and the other with a cup head. A form of socketed grub screw, requiring a special wrench, is shown in fig. 8. Standard sizes of all grub screws are given in B.S. 768.

**Special Bolts.**—Many bolts of special form are shown opposite and elsewhere in the pages of this book. The **tee-headed bolt,** fig. 9, is used to fasten work to slotted tables or plates. **Coupling bolts** are usually provided with cheese heads as in fig. 10; the "snug" is a pin, screwed or driven into the bolt, and fitting into a recess in the coupling to prevent the bolt from turning.

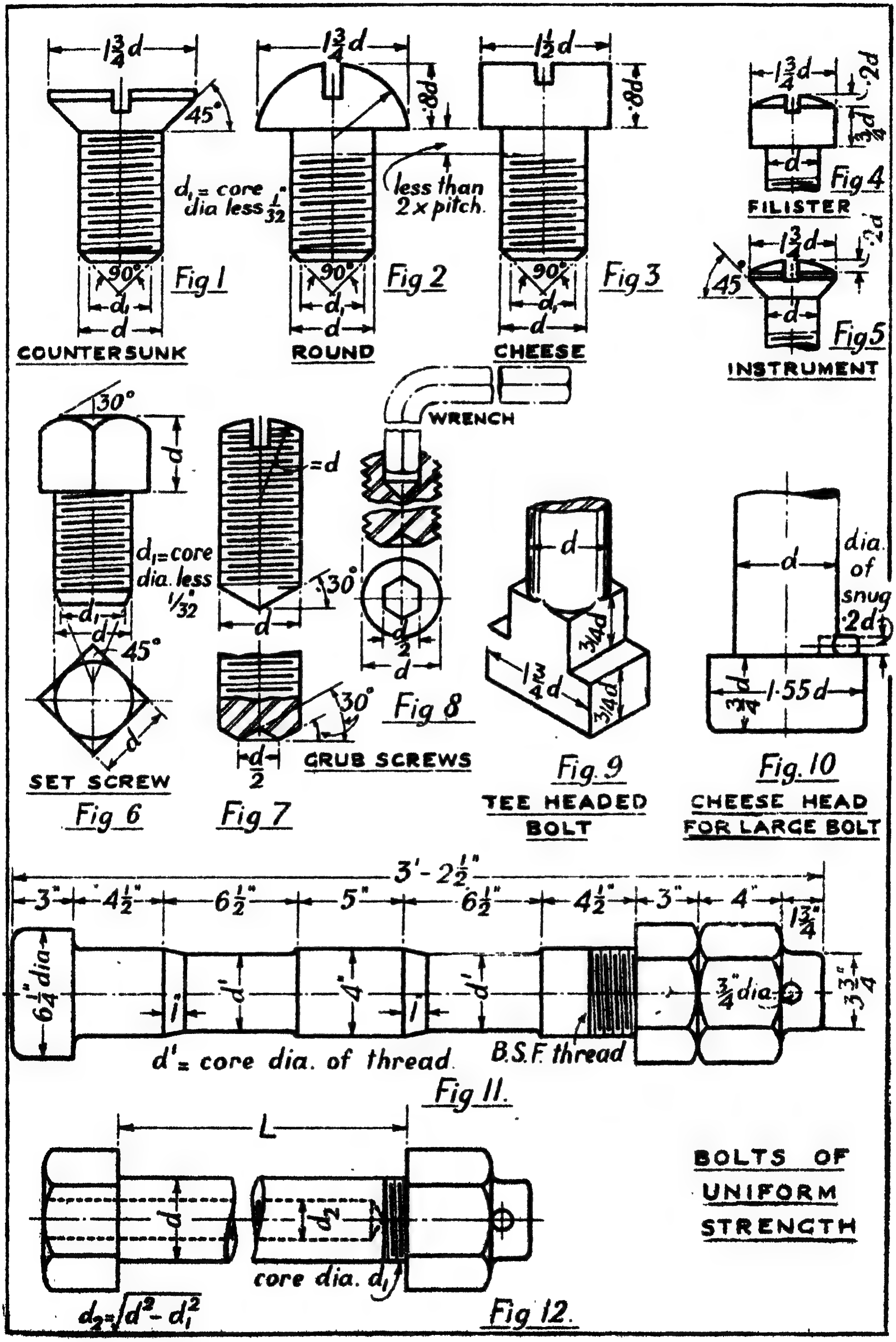
**Bolts of Uniform Strength.**—In an ordinary bolt the effect of impulsive loads applied axially is concentrated on the weakest part of the bolt, i.e. the cross-sectional areas at the roots of the threads. These weak sections are of small length and have a tendency to fracture if the stress is high. If, however, a portion of the unscrewed part is made equal in area to the area at the root of the threaded portion, a greater total stretch is permissible and the bolt is made both stronger and lighter. The bolt shown in fig. 11, which is suitable for the connecting rod shown on page 194, has a length of 11″ reduced and made equal to the core diameter. To facilitate entry into the connecting rod, the reduced lengths are coned at one end. A less common alternative is shown in fig. 12, in which an axial hole is drilled through the head as far as the threaded portion.

## EXERCISES

*Use the standard proportions for nuts and threads given on pp. 223 and 225*

(1) Draw, quarter size, the view of the bolt given in fig. 11, and add both a plan and an end view from right to left. If the U.T.S. of the material is 70,000 lb./in.², what is the maximum safe axial load for the bolt, assuming a factor of safety of 6?
*Answer.*—126,000 *lb. approx.*

(2) A coupling bolt 4″ dia., screwed B.S.Whit. thread, is of the type shown in fig. 12, L being 12″. Draw three views of the bolt and dimension them.


$1\frac{3}{4}d$
45°
$d_1$ = core dia less $\frac{1}{32}$"
90°
$d_1$
$d$
Fig 1
COUNTERSUNK
$1\frac{3}{4}d$
·8d
less than 2 x pitch.
Fig 2
ROUND
$1\frac{1}{2}d$
·8d
Fig 3
CHEESE
$1\frac{3}{4}d$
·2d
$\frac{3}{4}d$
Fig 4
FILISTER
45
Fig 5
INSTRUMENT
30°
$d_1$=core dia. less 1/32"
45°
SET SCREW
Fig 6
=d
30°
$\frac{d}{2}$
Fig 7
WRENCH
GRUB SCREWS
Fig 8
3/4d
$1\frac{3}{4}d$
Fig. 9
TEE HEADED BOLT
dia. of snug ·2d
1·55d
Fig. 10
CHEESE HEAD FOR LARGE BOLT
3'- $2\frac{1}{2}$"
3"
$4\frac{1}{2}$"
$6\frac{1}{2}$"
5"
4"
$1\frac{3}{4}$"
$6\frac{1}{4}$" dia
$d'$
$\frac{3}{4}$" dia
$3\frac{3}{4}$"
d' = core dia. of thread
B.S.F. thread
Fig 11.
L
$d_2$
core dia. $d_1$
$d_2 = \sqrt{d^2 - d_1^2}$
Fig 12.
BOLTS OF UNIFORM STRENGTH

**Strength of Bolts.**—If a bolt is not initially stressed, the maximum safe axial load which may be applied to it is given by the product (cross-sectional area at bottom of thread × permissible stress). An average working stress for steel is 10,000 lb./in.².

If, however, a bolt is first tightened up, the *initial* stress may be much greater than the additional stress produced by a load subsequently applied.* The initial stress caused by tightening the nut is indeterminate, and in fixing bolt diameters under these conditions practical experience is the only safe guide.

**Flanged Joints** are used for pipes, cylinder heads, &c. The total initial load applied to the bolts must be at least equal to the internal force tending to separate the flanges. In design work it is usual to assume that the greatest load on the bolts will occur when the joint is on the point of leaking, and that they are then called upon to resist a force equal to the full internal working pressure acting over the area of a circle just touching the inner sides of the bolt holes. (Possible bolt loads are discussed on pp. 56 and 57.) Values of $f$, the maximum permissible root stress in the bolts, are given below; they are progressively lower as the bolt diameter diminishes to allow for the unknown initial stress. Lower values still should be taken in rough work.

**Pipe Flanges.**—These have now been standardized—refer to Tables 7 and 8, pages 226, 227. In all joints the number of bolts is a multiple of 4 and the minimum bolt diameter is $\frac{1}{2}''$. For the joint shown in fig. 1, which is suitable for 2″ bore C.I. pipes and a working steam pressure of 150 lb./in.², we have:—

Assumed Load on Joint

$$= 150 \times \tfrac{1}{4}\pi \times (4\tfrac{5}{16})^2 = 2186 \text{ lb.}$$

Total resistance of Bolts

$$= 4 \times \text{root area of one bolt} \times f$$
$$= 0{\cdot}8128\,f.$$

Equating these, $\quad 0{\cdot}8128\,f = 2186.$

Hence the bolt stress

$$f = 2689 \text{ lb./in.}^2.$$

It will be evident that either fewer bolts or bolts of smaller diameter could be used while still keeping the stress within the safe value of 4000 lb./in.² given below.

**Joints for Cylinder Covers.**—Usually a bolt diameter is first assumed; the number of bolts is then calculated and the spacing checked. As an example, consider the studs of the cover in fig. 2. The cover is "spigoted" into the cylinder and, if the fit is a good one, it may be assumed that the steam pressure acts only over the cross-sectional area of the cylinder. Assuming studs 1″ in dia.:

Load on Cover = area of cylinder × maximum working pressure

$$= \tfrac{1}{4}\pi(18\tfrac{1}{2})^2 \times 250$$
$$= 67{,}200 \text{ lb.}$$

Resistance of Studs = no. of studs × root area × $f$ = N × 0·554 × 7600.

Equating these:

$$N = 67{,}200 \div (7600 \times 0{\cdot}554)$$
$$= 16, \text{ very nearly.}$$

The distance between the studs (pitch circle circumference ÷ 16) = $4\frac{1}{2}''$, and this is satisfactory for studs 1″ dia.

| Bolt dia. in inches | $\frac{1}{2}$ | $\frac{5}{8}$ | $\frac{3}{4}$ | $\frac{7}{8}$ | 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $1\frac{3}{8}$ | $1\frac{1}{2}$ | $1\frac{3}{4}$ |
|---|---|---|---|---|---|---|---|---|---|---|
| Stress $f$, in lb./in.² for steel | 3000 | 4000 | 5000 | 7000 | 7600 | 7900 | 8200 | 8500 | 8500 | 8500 |
| Approx. total load for B.S.Whit. threads | 360 | 800 | 1500 | 2950 | 4200 | 5400 | 7300 | 9000 | 11,000 | 15,000 |

## EXERCISES

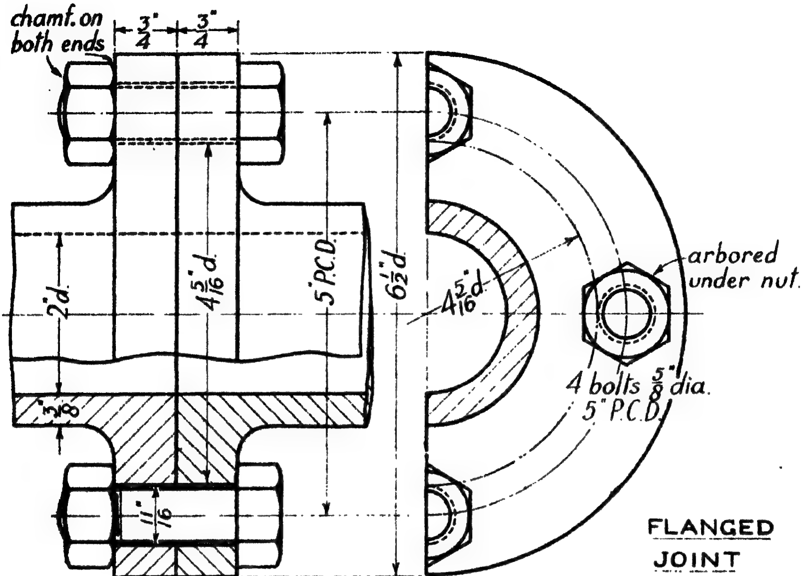
(1) If the B.S. eyebolt in fig. 3 is not initially stressed, determine a suitable dia. of screw for a direct load of 3 tons and prepare, full size, three dimensioned views of the eyebolt, using the proportions given.

X varies from $\frac{1}{16}''$ for $\frac{5}{8}''$, to $\frac{1}{8}''$ for 3″, eyebolts.
Y varies from $\frac{1}{8}''$ for $\frac{5}{8}''$, to $\frac{7}{16}''$ for 3″; eyebolts.

(2) Draw and dimension full-size views, arranged as in fig. 1, of a suitable C.I. flanged joint for a pipe 4″ dia., working steam pressure 100 lb./in.². Use Table 7, p. 226. Thickness of pipe $\frac{7}{16}''$.

(3) Determine a suitable number of studs $\frac{7}{8}''$ dia. for the cylinder cover shown in fig. 4, and prepare a working drawing of the cover, scale half size. Working pressure 350 lb./in.².

* It has been shown experimentally that the stress produced at the root section of a $\frac{3}{4}''$ dia. bolt when tightened up in the manner customary for pipe joints may be as high as 49,000 lb./in.² (Unwin).

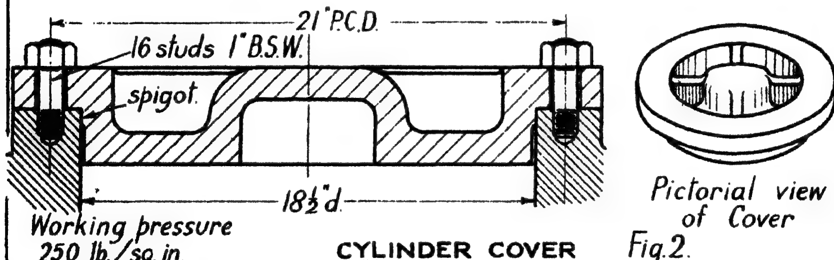



ELEVATION PART IN SECTION

Fig. 1.

PART END VIEW

FLANGED JOINT FOR PIPES.




Pictorial view of Cover

CYLINDER COVER

Fig. 2.

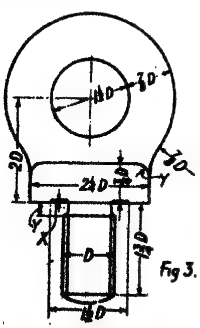



Fig 3.

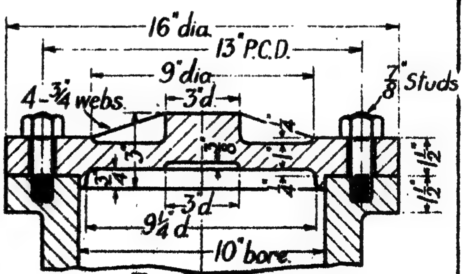



Fig. 4

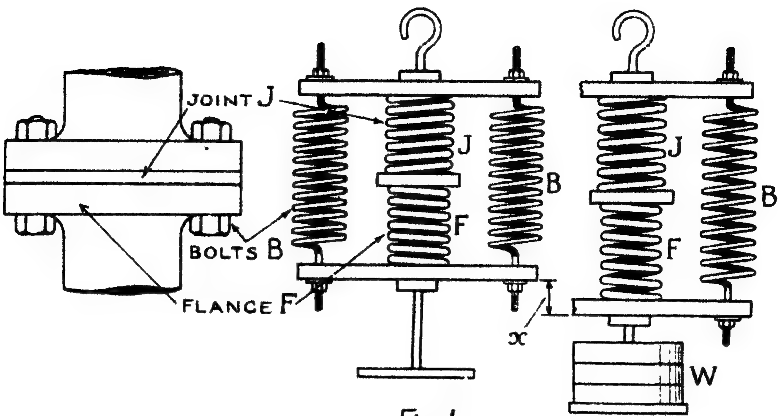



Fig 1

**Loads carried by Bolts** of flanged joints for pipes under fluid pressure.

Bolted flange joints are in such common use that it is of interest to consider the factors that govern the extra loads carried by the bolts (already tightened up) when fluid pressure is applied. It is assumed here that neither bolts nor jointing materials are strained beyond the elastic limit, and that there is negligible springing in the flanges between the bolt holes: probably neither assumption is justified. The treatment, however, gives the order of load carried by the bolts in various circumstances, and shows why bolt failure is a rare occurrence.

The bolted joint may be represented by the spring arrangement shown diagrammatically in fig. 1.

B is a stiff spring representing the bolts. Let its stiffness * be $s_b$.

F is a stiff spring representing the flanges. Let its stiffness be $s_f$.

J is a weaker spring representing the joint. Let its stiffness be $s_j$.

For convenience, suppose that only one spring B is used (in which case it could be within the springs J and F in the apparatus).

Now suppose the spring B to be put in tension by tightening the nuts, so that F and J are compressed. This condition represents the flanged joint with the bolts tightened. It is required to find the tension in B as the result of now applying a load W, representing the fluid pressure applied to the joint. Suppose all the springs to carry an initial load L, i.e. before the additional load W is applied.

Let the application of W cause a deformation $x$ (see figure); i.e. B elongates $x$, and F and J together elongate $x$.

Let P represent the added load in B, and Q the relieved load in F and J, as in fig. 2.

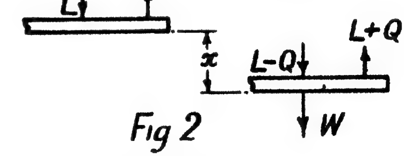



Fig 2

Hence $P + Q = W.$ . . . . . (1)

$P = x \times s_b.$ . . . . . . (2)

Now $\dfrac{\text{Extension of F}}{\text{Extension of J}} = \dfrac{s_j}{s_f};$ and

Extension of F + Extension of J $= x.$

* Stiffness is defined as the load causing unit deformation

$\therefore \frac{s_j}{s_f}$ (Extension of J)

+ Extension of J $= x$,

i.e. Extension of $J\left(\frac{s_j}{s_f} + 1\right) = x$. (3)

But Extension of J $\times s_j = Q$.
Substitution in (3) gives

$$Q = x \cdot \frac{s_j \times s_f}{s_j + s_f}. \quad . \quad . \quad (4)$$

Now $W = P + Q$

$$= x \cdot s_b + x \cdot \frac{s_j \cdot s_f}{s_j + s_f}.$$

From which

$$x = \frac{W}{s_b + \frac{s_j \cdot s_f}{s_j + s_f}}.$$

The added load $P = x \cdot s_b$.

$$\therefore P = \frac{W \cdot s_b}{s_b + \frac{s_j \cdot s_f}{s_j + s_f}},$$

or in more usable form,

$$P = \frac{W}{1 + \frac{s_f}{s_b}\left(\frac{1}{1 + \frac{s_f}{s_j}}\right)}. \quad . \quad . \quad (5)$$

Evidently, for springs all of the same proportions and material, $P = \frac{2}{3}W$. For the component parts of a flanged joint, stressed within the elastic limit, the stiffness varies directly as the modulus E, directly as the area $a$, and inversely as the length (or thickness) $l$.

Using the suffixes $f$, $b$, and $j$ for flange, bolt, and joint, as before, and assuming values of E and $l$ to be the same for bolts and flanges, then we have

$$\frac{s_f}{s_b} = \frac{a_f}{a_b}, \quad . \quad . \quad . \quad . \quad (6)$$

and

$$\frac{s_f}{s_j} = \frac{E_f}{E_j} \times \frac{l_j}{l_f}. \quad . \quad . \quad (7)$$

**Applications.**—

Ex. 1.—Thin metallic joint. Assume $\frac{E_f}{E_j} = 2$; $\frac{l_f}{l_j} = 16$; $\frac{a_f}{a_b} = 10$.

Substitution in (6), (7) and (5) gives $P = 0{\cdot}1\ W$.

Ex. 2.—As in Ex. 1, but taking $\frac{a_f}{a_b} = 4$. Then $P = 0{\cdot}22\ W$.

Ex. 3.—Easily compressible joint. Assume $\frac{E_f}{E_j} = 90$; $\frac{l_f}{l_j} = 10$; $\frac{a_f}{a_b} = 10$.

Substitution gives $P = 0{\cdot}5\ W$.

Ex. 4.—As in Ex. 3, but $\frac{a_f}{a_b} = 4$. Then $P = 0{\cdot}7\ W$.

The student should work out other combinations.

**Conclusion.**—The extra load on the bolts of a flanged joint, due to fluid pressure applied after tightening-up, is small when the flange area is large compared with the bolt area, and the jointing material is not easily compressible; a reduction in flange area has the effect of increasing this extra load.

If, however, an easily compressible joint is used, the extra load on the bolts may be an appreciable fraction of the total load due to fluid pressure.

**Reference.**—The student is referred to the Reports of the Pipe Flanges Research Committee of the Inst. Mech. E. (first report, 1936) for information on the relative efficacy of jointing materials, and the general behaviour of joints under conditions of high temperature and pressure.

**Steel and W.I. Piping.**—This may be (1) solid-drawn, (2) lap-welded, (3) butt-welded, or (4) riveted. The bulk of the piping used for pressure purposes is lap-welded, but for high-pressure steam solid-drawn piping is preferable.

If the thickness of the piping is calculated from the usual "thin cylinder" formula, $t = p \cdot d \div 2f$ (*vide* page 42) it will be found to be too small to give the rigidity required for practical purposes, especially if the working pressure is low. Empirical formulæ of the form $t = \frac{p \cdot d}{2f} + c$ are therefore used. For example, Board of Trade rules are:—

For solid-drawn steam pipes

$$t = \frac{p \cdot d}{2 \times 6000} + 0{\cdot}1''. \quad . \quad . \quad (1)$$

For welded steam pipes

$$t = \frac{p \cdot d}{2 \times 4500} + 0{\cdot}12''. \quad . \quad (2)$$

**British Standard Pipe Threads.**—These are of the Whitworth *form*, and full details of the pitch, length of thread, &c., are given in Table 5, page 225. It should be noted that 11 threads per inch are used for all pipes from 1" to 6" "nominal bore". The outside diameter of a tube corresponds approximately to the dimensions given in the table, but the actual bore will vary with the working pressure and may not be equal to the nominal bore given.

**Screwed Tubes and Couplers. Figs. 1 and 2.**—The screw thread on the pipe end may be either parallel to the pipe axis or coned $\frac{1}{16}''$ on the diameter per inch. of length. The common form of coupler has a parallel thread, and is screwed on to a coned pipe end. The gauge diameter of the tapered screw and its position are defined in the table.

Details of tee-pieces, elbows, reducing pieces, and other fittings, for steel and W.I. screwed piping, are not included here. The fittings have been standardized, and particulars are given in Report No. 154 of the B.S.I.

**Screwed-on Flanges for Steam Pipes.***—A suitable flange for pipes subjected to steam pressures up to 250 lb./in.$^2$ is shown in fig. 3. The flange is first screwed with a parallel full thread. A portion of the thread is then cut off to a taper, leaving a short parallel plain part within the boss. The dies with which the pipe thread is cut are ground off to a corresponding taper. When the flange is in place, the end of the pipe is expanded into the recess at the face of the flange. This type of flange is provided for pipes up to 10" bore.

A type of flange for steam of a working pressure of 550 lb./in.$^2$ is shown in fig. 4. The end of the thread at the face of the flange is sealed by electric welding. Slight bosses on the flange faces are accurately machined and scraped for a *metal to metal* joint. The shanks of the bolts are reduced in order that the danger of fracture at the thread roots under shock loads may be eliminated.

In both designs, figs. 3 and 4, the face of each flange is relieved at the bolt holes by a shallow annular recess.

The design of special joints for the

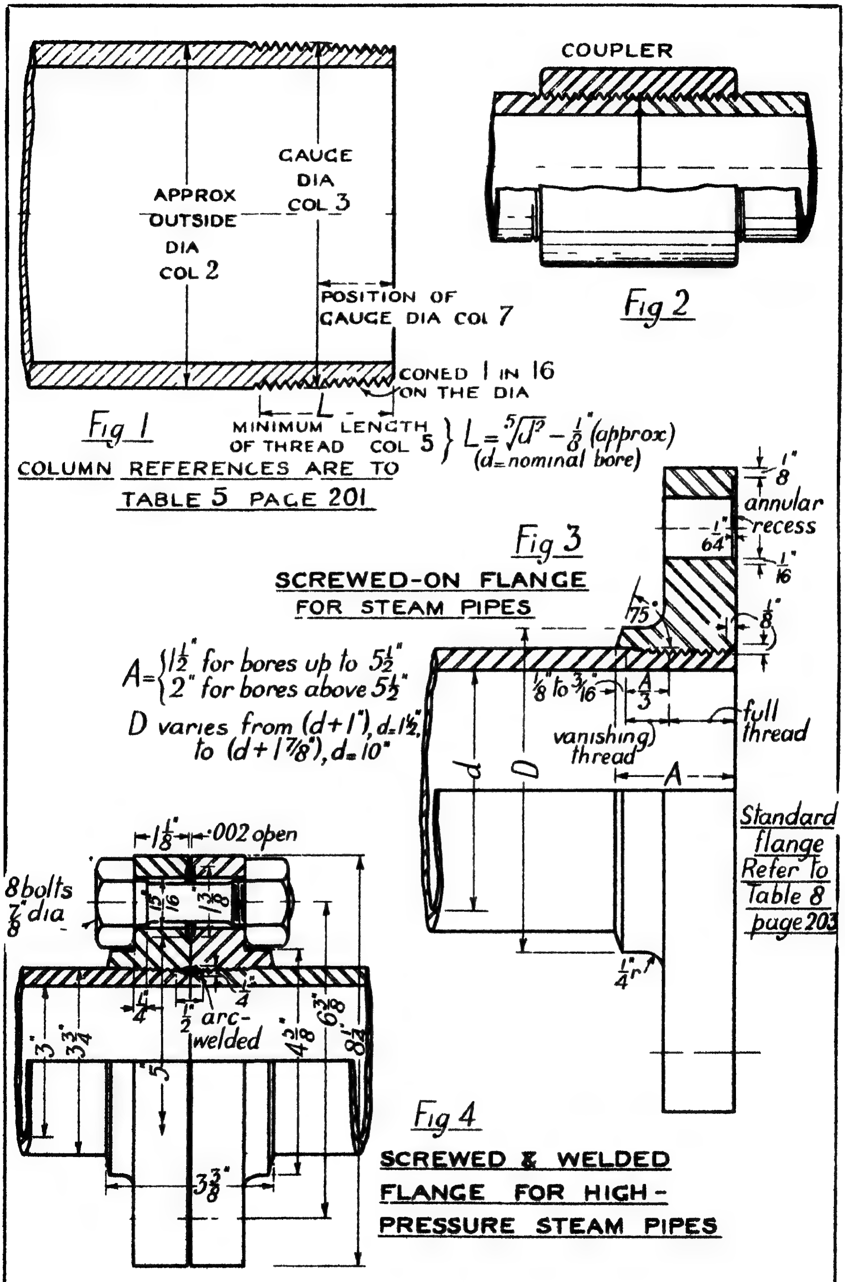
## EXERCISES

(1) Using Table 8, together with the proportions given in fig. 3, prepare, full size, the following dimensioned views of a screwed flanged joint for a 3" bore steel pipe, working pressure 250 lb./in.$^2$, showing the two flanges bolted together: elevation, upper half in section through a bolt; end view.

(2) Prepare views similar to those required in Exercise (1) of the joint shown in fig. 4. Calculate the stress in the flange bolts if the joint is on the point of leaking and the steam is acting over the area of a circle 5" dia.

*Answer.*—3200 *lb./in.*$^2$.

* Where weight is an important factor, as in naval machinery, the flange proportions given in Tables 7 and 8, pp. 226 and 227, are unsuitable and lighter scantlings are used.




$L = \sqrt[5]{d^2} - \frac{1}{8}$ (approx) ($d$ = nominal bore)

$A = \begin{cases} 1\frac{1}{2}" \text{ for bores up to } 5\frac{1}{2}" \\ 2" \text{ for bores above } 5\frac{1}{2}" \end{cases}$

$D$ varies from $(d+1")$, $d = 1\frac{1}{2}"$, to $(d+1\frac{7}{8}")$, $d = 10"$

very high temperatures and pressures (930° F., 1850 lb./in.$^2$) now being adopted has been the subject of much research and experiment.

**Thick Pipes and Cylinders.**—When internal fluid pressure is applied to a cylinder whose walls are thick in relation to its diameter, the inner layers are stressed much more highly than the outer, and account must be taken of the *maximum* stress produced, not the average over the section (as for thin cylinders). The following formula * is obtained by Lamé's theory:

$$t = \frac{d}{2}\left\{\sqrt{\frac{f+p}{f-p}} - 1\right\}, \quad . \quad (1)$$

where $t$ = thickness in inches, $d$ = internal diameter in inches, $p$ = fluid pressure in lb./in.$^2$, and $f$ = maximum safe principal stress † in lb./in.$^2$. When $p$ represents the *test* pressure, $f$ should not exceed 9000 for C.I. and 25,000 for steel castings.

**C.I. Pipes and Flanges for Hydraulic Power.**—Two pressure ranges are in common use for hydraulic power: (*a*) for working pressures from 700 to 900 lb./in.$^2$, test pressure 2500 lb./in.$^2$; (*b*) for working pressures from 900 to 1200 lb./in.$^2$, test pressure 3300 lb./in.$^2$. Higher pressures are used for special purposes.

The flanges are elliptical in shape and require two bolts only. A circular projection on one flange fits into a recess in the other and bears against a gutta-percha ring of tapering section. The bolts are square-necked and the holes are square, ample clearance being allowed between bolt and hole. The joint permits a certain amount of flexibility in the pipe line, and this is probably an important factor in its success.

Two flange types are shown. Type 2 is more robust in design than Type 1, and is used for larger sizes of pipes. A stronger coupling results if the flanges are set well back on the pipe, as in Type 2 (dimension M).

The pipe thickness may be calculated from the above formula, and the bolt diameter settled as on p. 54. The general proportions, however, are based largely on successful practice, and they have been standardized for pipe bores of 2″, 3″, 4″, 5″, 6″, 7″, and 8″; detailed dimensions are given in Table 9, p. 228, reference being made there to the letters in the drawings. Dimensions carrying no index letter are common to all sizes.

A dimensioned drawing of a **tee-piece** is given on p. 171: the student should refer to this drawing, which shows a slightly different view of the flanges from that given opposite.

Details of a low-pressure C.I. **heating pipe** and socket joint are given on p. 170.

## EXERCISES

(Show the flanges bolted together with the faces $\frac{1}{16}''$ apart. Use square-headed bolts and hexagonal nuts. Dimension the views.)

(1) Draw, full size, the following views of the Type 1 flanges shown: elevation, with the upper half in section through the bolt holes; plan; end view on the face of each flange.

(2) Draw, half size, the following views of the Type 2 flanges shown: elevation, with the upper half in section through the bolt holes; plan; end view on the nuts.

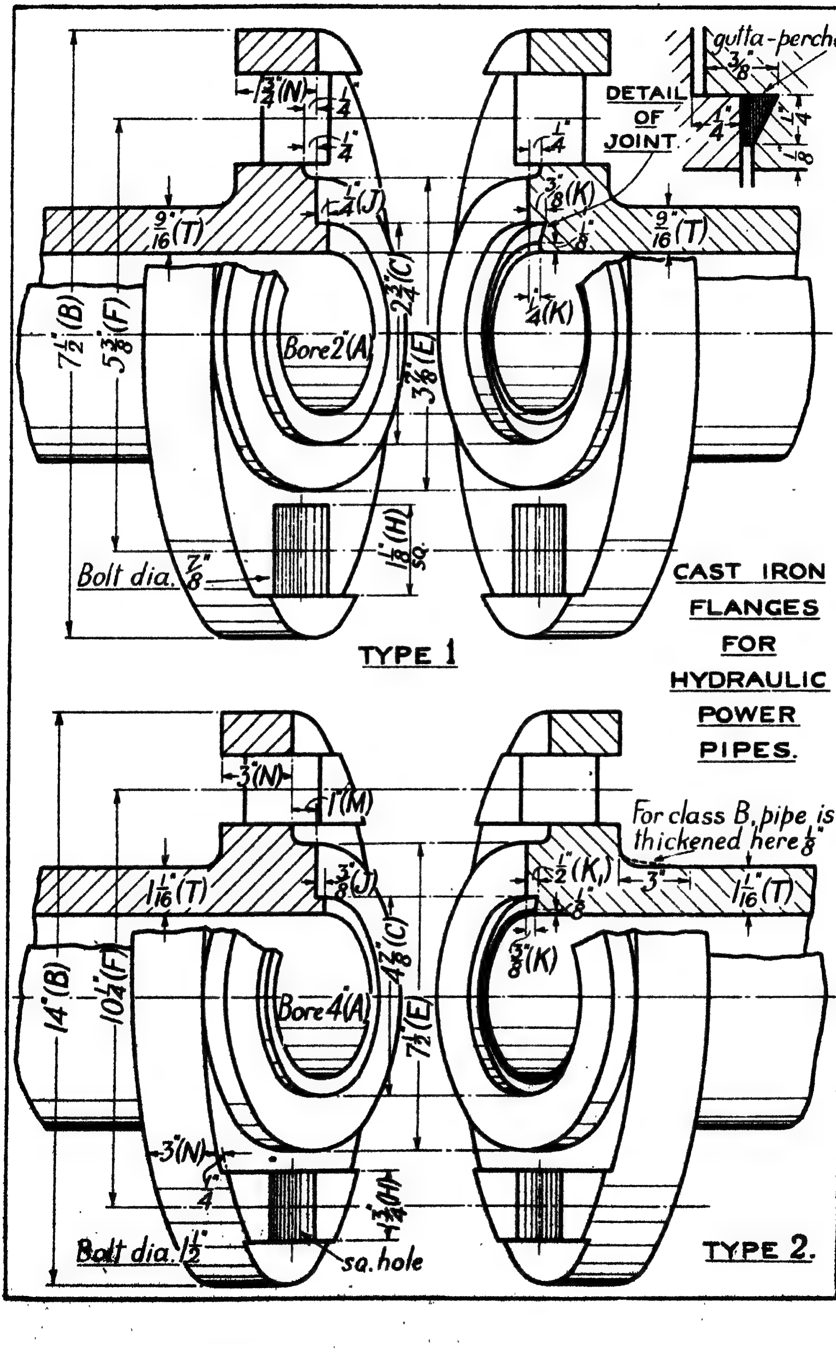
(3) Calculate, for the joints shown, (*a*) the maximum stress in each of the pipes, (*b*) the stress in the bolts, taking test pressures of 2500 for Type 1 and 3300 for Type 2.

*Answers.*—(*a*) 5970; 8210 *lb./in.*$^2$. (*b*) 17,610; 23,670 *lb./in.*$^2$.

* An alternative formula deduced from the maximum principal strain is

$$t = \frac{d}{2}\left\{\sqrt{\frac{3f+2p}{3f-4p}} - 1\right\}.$$

† i.e. the circumferential or hoop tension at the inner surface; there is also a radial compressive stress.


DETAIL OF JOINT
gutta-percha
Bore 2" (A)
Bolt dia. 7/8"
TYPE 1
CAST IRON FLANGES FOR HYDRAULIC POWER PIPES.
For class B, pipe is thickened here 1/8"
Bore 4" (A)
Bolt dia. 1 1/2"
sq. hole
TYPE 2.

**Expansion of Steel and Cast-iron Pipes.**—Some provision must be made in long pipe lines to accommodate the alteration in length occasioned by temperature changes. High-pressure steam, having a temperature of 750° F. and over, is now in common use: the alteration in length of a steel pipe line 40′ 0″ long for a temperature variation of 700° F. is $480 \times 7{\cdot}9 \times 10^{-6} \times 700 = 2{\cdot}65''$, taking $7{\cdot}9 \times 10^{-6}$ as the average value of the coefficient of linear expansion for steel per degree F. between 50° F. and 750° F. A common expansion allowance for C.I. heating mains is 1″ per 40′ 0″: this allows a variation in temperature of 300° F., taking the coefficient of expansion as $6{\cdot}4 \times 10^{-6}$. The change of length is taken up usually either by expansion bends, bellows pieces, or by sliding joints.

**Expansion Bends.**—Two types are shown opposite, the "lyre" type being preferable to the U. The mean radii R of the bends should not be less than $5d$, where $d$ is the pipe bore. The length of the straight portion $a$ varies from $d$ to $2d$.

A reaction of considerable magnitude is produced at the anchorage for an expansion bend, and its value, together with that of the maximum stress * set up in the material at the bend, are given by the following formulæ:

$$L = C\frac{F \cdot R^3}{E \cdot I}, \quad . \quad . \quad . \quad (1)$$

$$f = c\frac{d_1 \cdot L \cdot E}{R^2}, \quad . \quad . \quad . \quad (2)$$

where L = axial change of length in inches, $f$ = stress in lb./in.², F = anchoring force in pounds, R = mean radius of bend in inches, $d_1$ = outside diameter of pipe in inches, E = Young's modulus in lb./in.², I = moment of inertia of the pipe section in inch units.

C and $c$ are constants having the following values:

Expansion U bend:

$C = 9{\cdot}425; \quad c = 0{\cdot}106.$

Lyre-shaped bend:

$C = 39{\cdot}888; \quad c = 0{\cdot}0427.$

**Expansion Joints.**—A simple type of expansion joint is shown opposite. The sleeve slides in a socket, and leakage is prevented by an asbestos-packed stuffing-box. The sleeve is allowed a total movement of 2½″, the two stay bolts being designed to prevent further withdrawal when the sleeve is still within the socket ½″. The stay bolts are screwed into projections on flange A and are riveted over; the keep-nuts at the other end are secured by ⅛″ split pins. All parts are cast in gun-metal; the bolts, studs, and nuts are of steel. Large joints are usually of C.I. or C.S. with the sliding parts bushed with G.M. An example of this type is given on pp. 178 and 179.

The anchoring force necessary for an expansion joint is indeterminate, and depends largely upon the alignment of the pipe line and the manner in which the joint is packed.

## EXERCISES

(1) Draw, half size, the following dimensioned views of the expansion joint given: elevation, half in section as in figure; plan; end view from right to left. Show the sleeve 1″ out and determine suitable lengths for the gland studs and stay bolts. Minor dimensions have been omitted purposely in the figure.

(2) A steam main, 6″ nominal bore, 40′ 0″ long, carries steam at a pressure of 450 lb./in.² and has a temperature variation of 435° F. Make a scale working drawing of an expansion bend of the "lyre" type and calculate ($a$) the anchoring force required, ($b$) the stress in the material. Take $d_1$ = 6⅝″, R = 30″, I = 27 in.⁴, E = $29 \times 10^6$ lb./in.².

*Answers.*—($a$) 1200 *lb.* ($b$) 14,800 *lb./in.² approx.*

* The maximum stress may be reduced by putting the pipe line in tension when cold, preferably so that the bend is opened by half the expected movement.

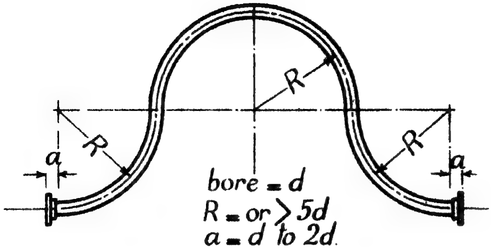



Fig. 1.
EXPANSION U - BEND

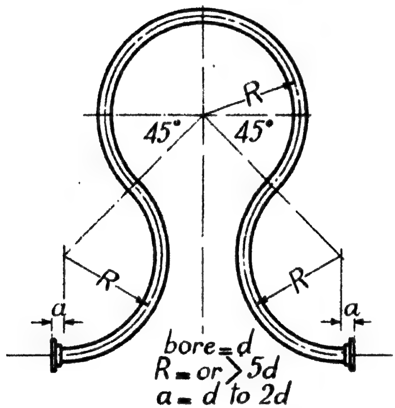



Fig. 2.
LYRE SHAPED BEND

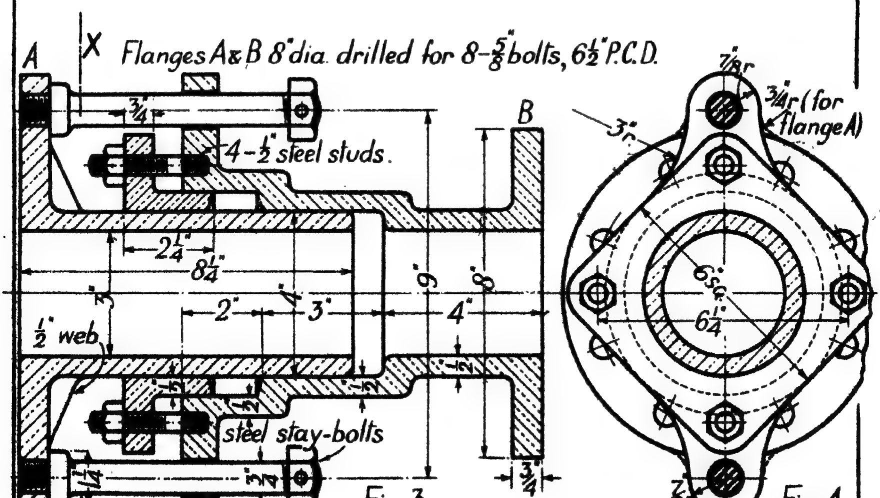



Fig. 3.
SECTIONAL ELEVATION

Fig. 4.
SECTIONAL END VIEW ON XX.

3" BORE GUNMETAL EXPANSION JOINT.

**Knuckle or Pin Joints** are used to connect two or more rods whose axes meet in a point, the point of intersection being on the axis of the pin. Typical examples in which knuckle joints are used as connexions are: ties for roof trusses; the links of a suspension chain; machinery parts in which one rod is to have a small relative motion about another, e.g. valve rods.

The joint has many forms, one of the most common being that shown in fig. 1. The rod ends are forged to shape, one being forked and the other provided with an eye to fit within the jaws of the fork. The pin of the joint passes through both eyepiece and fork, and may be secured either by a collar and tapered pin (taper $\frac{1}{4}''$ per foot on the diameter), fig. *2a*, or by a thin nut screwed up to a shoulder on the end of the pin, fig. *2b*. The pin is prevented from rotating in the fork by means of a small stop pin or peg.

In small pin joints one eye of the forked end is often screwed to take a thread provided on the end of the pin: the head of the pin may be of the countersunk type.

In better-class work the sides of the fork and eyepiece are machined, the hole is accurately bored, and the pin is turned.

**Proportions.**—These are largely empirical and are given in figs. 1 and 2, the unit being the diameter of the rods to be connected: it is assumed that all parts are of the same material, wrought iron or steel. If the joint is accurately made and is not slack, the pin will be in a condition of double shear. Let the joint be subjected to an axial pull P, and let D = diameter of the rods to be connected, and $D_1$ = diameter of the pin; let $f_t$ and $f_s$ be the tensile and shear tresses.

Then, for the rods,

$$P = \tfrac{1}{4}\pi \,.\, D^2 \,.\, f_t,$$

and, for the pin,

$$P = 2(\tfrac{1}{4}\pi \,.\, D_1^2)f_s.$$

Equating these, for uniform strength,

$$\tfrac{1}{4}\pi \,.\, D^2 f_t = 2(\tfrac{1}{4}\pi \,.\, D_1^2)f_s,$$

$$\text{i.e.}\quad D_1^2 = \tfrac{1}{2}D^2 \,.\, f_t/f_s.$$

Assuming $f_s/f_t = 0{\cdot}7$, then

$$D_1^2 = D^2 \div 1{\cdot}4,$$

and $\qquad D_1 = {\cdot}84D$ approx.

Usually the parts are assembled with ample clearance, and the pin is subjected therefore to a small bending moment in addition to the shearing force. By making the diameter of the pin equal to that of the rods, as in fig. 1, a margin of strength is provided to allow for this additional stress, the value of which is not readily determined. (An approximate solution is given in *Machine Design* by Unwin and Mellanby.)

**Joint for Three Rods.**—The knuckle connexion is suitable for uniting more than two rods in a common joint, and fig. 3 shows three rod ends, two forked and one plain, having a common pin. The proportions given in figs. 1 and 2 require a little modification here; for example, the pin being longer should be slightly larger in diameter: the settlement of detailed dimensions is left as an exercise for the student (see Ex. 2).

## EXERCISES

(1) Draw, full size, the following views of a knuckle joint of the type shown in fig. 1 for rods 1″ dia.: elevation, plan, and sectional end view on axis of pin. Adopt the fastening shown in fig. *2b*. Dimension the views.

(2) Draw, half size, the following views of a knuckle joint to connect three rods, $1\frac{1}{4}''$ dia., as in fig. 3: elevation, plan, and end view. Give also a separate detail of the pin. Dimension the views.

## COTTERED JOINT FOR RODS

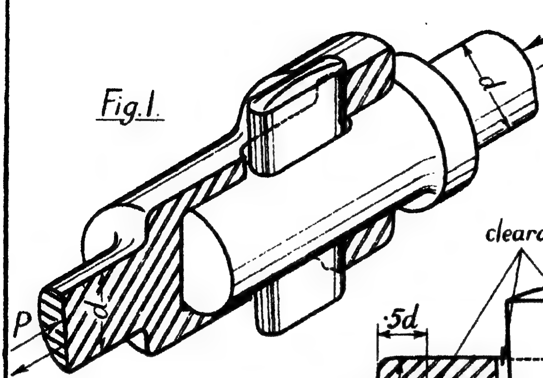



Fig. 1.

$D = 2{\cdot}4d \quad D_1 = 1{\cdot}75d$

$d_1 = 1{\cdot}21d \quad d_2 = 1{\cdot}5d$

$a = c = {\cdot}75d$

$b = 1{\cdot}3d \quad t = {\cdot}31d$

$t_1 = {\cdot}45d$

$P = \pi/4\, d^2 f_t$

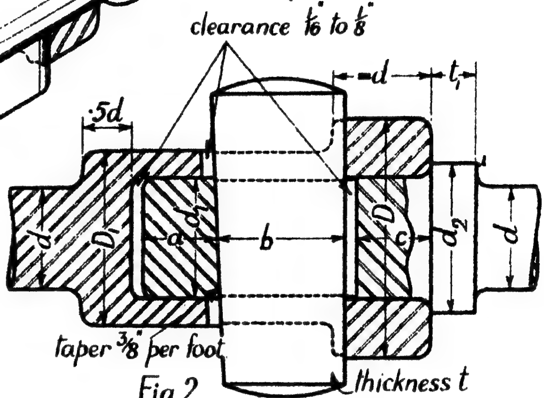



Fig. 2

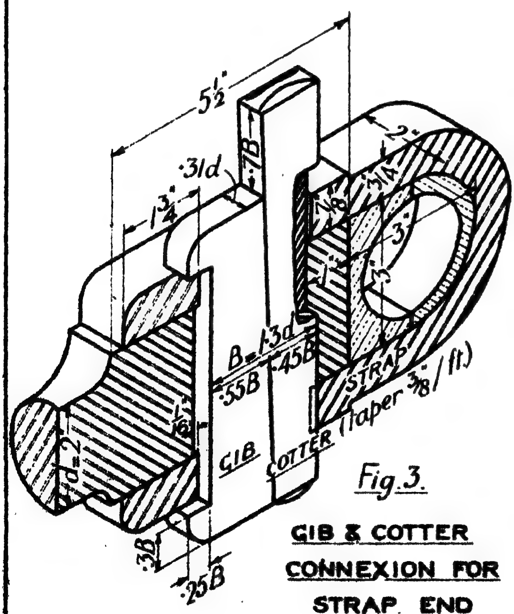



Fig. 3.

GIB & COTTER CONNEXION FOR STRAP END

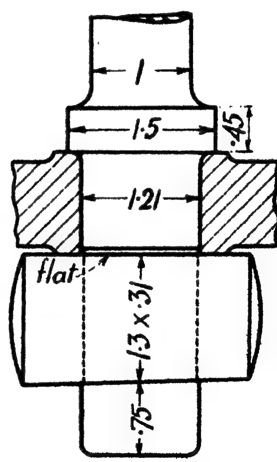



Fig. 4

COTTERED STANDARD

**Shafts transmitting power** are subjected principally to torsion, usually in conjunction with bending. Fig. 1 shows a line shaft S belt-driven by a motor M, and transmitting power to machine tools by means of pulleys and belting: the drive to the lathe L is transmitted through a countershaft CS, introduced chiefly to provide a means of speed variation. The line shaft is in lengths joined by couplings C, and is supported by bearings carried on brackets B.

### Strength of Circular Shafts.

Let $d$ = diameter of shaft (in.).
$T$ = twisting moment (lb. in.).
$M$ = bending moment (lb. in.).
$N$ = speed (revs./min.).
$f_s$ = maximum shear stress (lb./in.$^2$).
$f$ = maximum tensile or compressive stress (lb./in.$^2$).

Shaft subjected to **Simple Torsion**, fig. 2:

$$T = \pi \, . \, d^3 \, . \, f_s \div 16. \qquad (1)$$

Shaft subjected to **Simple Bending**, figs. 3 and 4:

$$M = \pi \, . \, d^3 \, . \, f \div 32. \qquad (2)$$

If the shaft is supported on both sides of the load, as is usually the case, the reactions at the supports must first be found and the greatest bending moment calculated.

Shaft subjected to **Combined Torsion and Bending**, figs. 5 and 6:

**Equivalent Twisting Moment,***

$$M_{et} = \sqrt{T^2 + M^2}, \quad . \quad . \quad . \qquad (3)$$

so that

$$\sqrt{T^2 + M^2} = \pi d^3 f_s \div 16. \qquad (4)$$

For **Hollow Shafts** replace $d^3$ in the above by $(D^4 - d^4) \div D$, where $D$ = external diameter, $d$ = internal diameter.

**The Horse-power** transmitted by shafting is given by:

$$H.P. = 2\pi \, . \, N \, . \, T \div (33{,}000 \times 12). \qquad (5)$$

Commonly accepted values for the stresses are: $f = 11{,}000$, $f_s = 9000$ lb./in.$^2$.

**Line Shafting.**—The bending action on line shafting due to the weights of pulleys and the tension in the belting connected with them is not easily estimated, and it is usual first to calculate a suitable shaft diameter to resist torsion only, and then to multiply this diameter by a factor to cover (*a*) the effect of other straining actions, (*b*) variations in the twisting moment. **The factor 1·34** is commonly used. The following are the **shaft speeds** usually adopted:—

Machine Shops, 150–200 revs./min.
Woodworking Shops, 250–300 revs./min.
Mills, 300–400 revs./min.

The distance $D_s$ in feet between the bearings is given by the empirical formula

$$D_s = 5\sqrt[3]{d^2}. \quad . \quad . \quad . \qquad (6)$$

## EXERCISES

(*Take* $f = 11{,}000$ *lb./in.*$^2$, $f_s = 9000$ *lb./in.*$^2$.)

(1) Determine the diameter of a solid circular shaft to transmit a twisting moment of 113,000 lb. in.

*Answer.*—4″ *approx.*

(2) Determine the diameter of a solid circular shaft to withstand a bending moment of 29,000 lb. in.

*Answer.*—3″ *approx.*

(3) Determine the diameter of a solid circular shaft to withstand *both* the twisting moment of (1) and the bending moment of (2).

*Answer.*—$4\frac{1}{16}$″ *approx.*

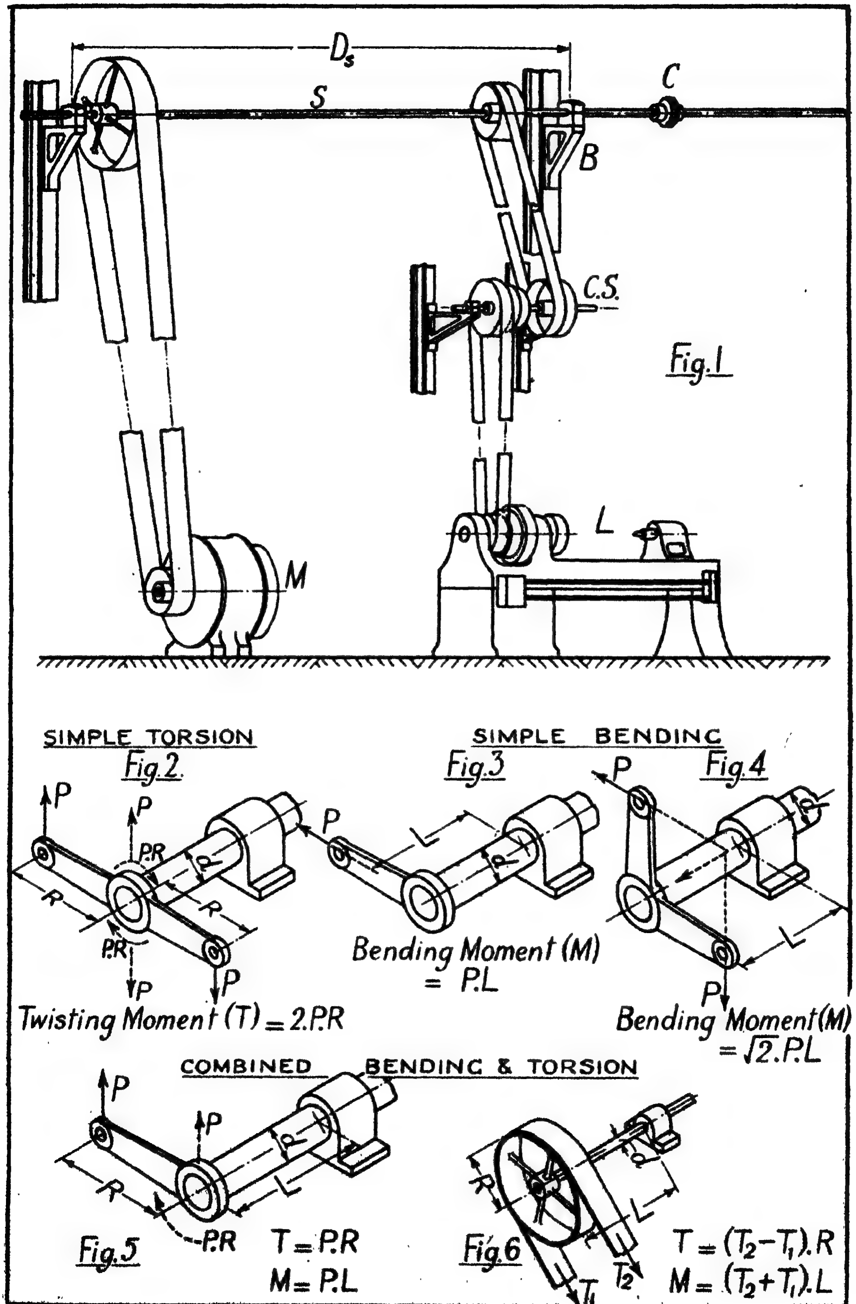
(4) Determine the diameter of a solid circular shaft to transmit 100 h.p. at 200 revs./min., allowing a factor of 1·34.

*Answer.*—$3\frac{1}{2}$″.

(5) If the shaft in (4) is replaced by a hollow shaft *of the same weight*, internal diameter 2″, what horse-power will it transmit at 200 revs./min.?

*Answer.*—143 *h.p. approx.*

* Guest's formula for ductile materials, based on the theory of greatest shearing stress.


Ds
S
C
B
C.S.
Fig.1
L
M
SIMPLE TORSION
Fig.2.
P
P
P.R
d
R
R
P.R
P
P
Twisting Moment (T) = 2.P.R
SIMPLE BENDING
Fig.3
P
L
d
Bending Moment (M)
= P.L
Fig.4
P
L
P
Bending Moment (M)
= √2.P.L
COMBINED BENDING & TORSION
P
P
R
d
L
P.R
Fig.5
T = P.R
M = P.L
R
d
L
Fig.6
T1
T2
T = (T2−T1).R
M = (T2+T1).L

**A Key** is a piece inserted between a shaft and a hub, usually in an axial direction, to prevent relative rotation. Keys are almost invariably made of steel. They may be divided into four classes: (1) sunk, (2) saddle, (3) tangent, (4) round.

**Sunk Keys.**—Four types are shown opposite: (*a*) rectangular, (*b*) gib-head, (*c*) feather, and (*d*) Woodruff.

(*a*) **The rectangular key,** fig. 1, is the most common. It is driven into a keyway cut half in the shaft and half in the hub, and should fit neatly at the sides and tightly at the top and bottom. Square keys are also used.

(*b*) **The gib-head key,** fig. 2, is the ordinary rectangular key with a head formed on one end to facilitate removal.

(*c*) **The feather key** is attached to one member of a pair and permits relative axial movement. Hence a working clearance is allowed in the sliding keyway at the top and the sides. The feather key is commonly fitted into a recess cut in the shaft, fig. 3, and secured by screws. The above types may be either parallel, or tapered in thickness 1 in 100.

(*d*) **The Woodruff key,** fig. 7, is a slice from a bar of segmental cross-section. It is used in machine tool and automobile construction. The key fits a recess milled in the shaft and adjusts itself to any taper in the keyway of the hub. It should not be used as a feather. The letters are referred to in the table on p. 228.

**Saddle Keys,** figs. 5 and 6, are suitable for light service. Flat saddle keys tend to rock on the shaft under heavy duty and work loose. Hollow saddle keys hold by friction and are useful as temporary fastenings.

**Tangent Keys** are fitted each to withstand torsion in one direction only. **Round keys** are of circular section and fit holes drilled partly in the shaft and partly in the hub (refer to pp. 87 and 89).

**Strength of Sunk Keys.**—Let $d$ = diameter of shaft in inches, $l$ = length, $b$ = breadth, $t$ = thickness of key, in inches; let $f_s$ = shear stress in shaft and $f_k$ = shear stress in key, both in lb./in.$^2$.

In practice $l = \text{or} > 1\frac{1}{2}d$.

Turning moment on shaft

$$= \pi \,.\, d^3 \,.\, f_s \div 16. \quad . \quad . \quad (1)$$

Shearing resistance of key

$$= l \,.\, b \,.\, f_k = 1\tfrac{1}{2}d \,.\, b \,.\, f_k.$$

Moment of shearing resistance about shaft axis

$$= \tfrac{1}{2}d(1\tfrac{1}{2}d \,.\, b \,.\, f_k). \quad . \quad . \quad (2)$$

Equating (1) and (2),

$$\tfrac{3}{4}d^2 \,.\, b \,.\, f_k = \pi \,.\, d^3 \,.\, f_s \div 16.$$

From which

$$b = \pi \,.\, f_s \,.\, d \div 12 f_k.$$

If $f_s/f_k = 0{\cdot}8$, then $b = 0{\cdot}2094d$.

In practice

$$b = \tfrac{1}{4}d \quad \text{and} \quad t = \tfrac{2}{3}b, \text{ approx.}$$

Let $f_c$ = crushing stress in lb./in.$^2$.

Bearing surface at side of keyway

$$= \tfrac{1}{2}t \,.\, l = \tfrac{1}{2} \,.\, \tfrac{2}{3} \,.\, b \,.\, l.$$

Equating the crushing resistance to the shear resistance of the key,

$$\tfrac{1}{3}b \,.\, l \,.\, f_c = b \,.\, l \,.\, f_k,$$

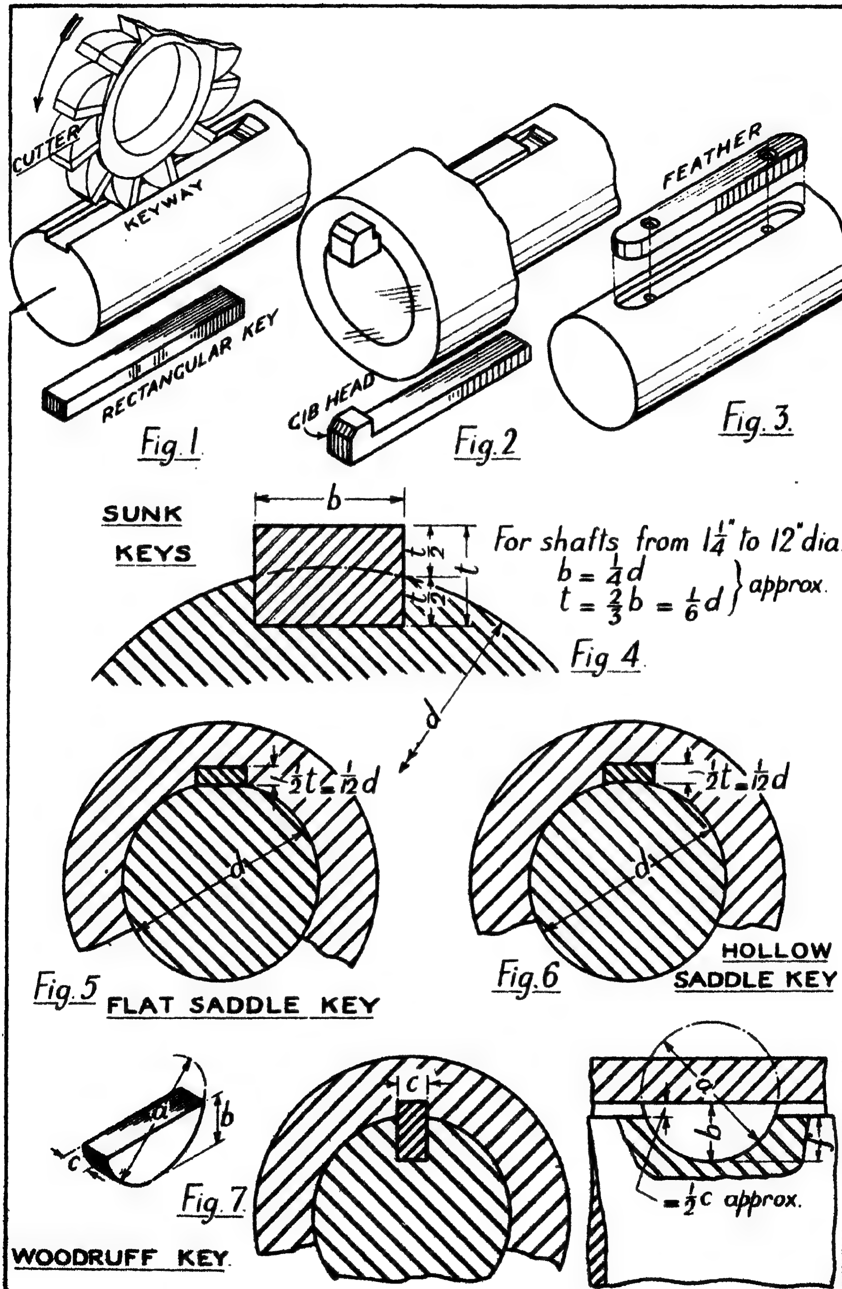
from which

$$f_c/f_k = 3.$$

Hence, neglecting the resistance due to radial compression, the crushing stress is three times as great as the shearing stress. The wedge action of a well-fitted key, however, reduces the crushing effect at the sides considerably, although no relief is given to feathers in this way.

**Spline Shafts** are widely used in automobile work in place of shafts with feathers. A spline shaft has a number of key-like projections integral with it, equally spaced round the circumference. These engage with corresponding recesses in a spline hub.

* Various types of keys, with their keyways, have been standardised, and extracts from the British Standard Specification No. 46, Part I, 1929, are given in Tables 10 and 11, pp. 228 and 229.


CUTTER
KEYWAY
RECTANGULAR KEY
Fig. 1.
GIB HEAD
Fig. 2
FEATHER
Fig. 3.
SUNK KEYS
b
For shafts from 1¼" to 12" dia.
b = ¼d
t = ⅔b = ⅙d
approx.
Fig 4.
d
½t = 1/12 d
Fig. 5 FLAT SADDLE KEY
Fig. 6
HOLLOW SADDLE KEY
a
b
c
Fig. 7
WOODRUFF KEY
= ½c approx.

**Cast-iron Split Muff Coupling. Fig. 1.**—The two shafts to be connected butt at the middle of the coupling and are firmly gripped when the halves of the coupling are bolted together. Keyways are cut in the shafts and a parallel key is used. The couplings are recessed for nuts and bolt heads, clearance being provided around the nuts for a box spanner. Slight shoulders are left on the recessed faces to prevent the bolts from working outwards under centrifugal force. The bolts on each side are inserted from opposite directions, for convenience when tightening up. Spring washers are fitted under the nuts.

Large muff couplings are usually cast hollow, as shown by the section given in fig. 2.

Suitable proportions for larger couplings are as follows:

| Shaft. Dia. | Coupling. Dia. | Coupling. Length. | Bolts. Dia. | Bolts. Length. |
|---|---|---|---|---|
| 3" | 7" | $10\frac{5}{8}$" | $\frac{5}{8}$" | $2\frac{9}{16}$" |
| $3\frac{1}{2}$" | 8" | $12\frac{3}{8}$" | $\frac{3}{4}$" | 3" |
| 4" | 9" | $14\frac{1}{8}$" | $\frac{7}{8}$" | $3\frac{1}{2}$" |
| $4\frac{1}{2}$" | $10\frac{1}{2}$" | $15\frac{3}{4}$" | 1" | $3\frac{7}{8}$" |

**Flanged Coupling. Fig. 3.**—The flanges, usually of C.I., are pressed on the shafts and keyed from the faces, the keys having a taper of $\frac{1}{8}$" per foot. The machined faces are held together by steel bolts which are a driving fit in their holes. Alignment is given, independently of the bolts, either by the provision of a turned projection on one flange which fits into a corresponding recess in the other, as shown, or by allowing the end of one shaft to enter for a short distance the coupling on the other. To guard against accident the flanges are formed to shroud the nuts and bolt heads.

**Proportions.**—The forces which may act on a coupling, due to errors in alignment and other causes, are to a large extent indeterminate, and the proportions adopted are based on empirical rules which cover designs found to be successful in practice. For shaft diameters from 2" to 6" the proportions given in fig. 3 are suitable: details of a coupling for a $1\frac{1}{2}$" dia. shaft are given on p. 109. The numbers of bolts and their diameters are as under:

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Shaft Dia. | 2" | $2\frac{1}{2}$" | 3" | $3\frac{1}{2}$" | 4" | 5" | 6" |
| Bolt Dia. | $\frac{1}{2}$" | $\frac{5}{8}$" | $\frac{3}{4}$" | $\frac{3}{4}$" | $\frac{7}{8}$" | $1\frac{1}{8}$" | $1\frac{1}{4}$" |
| No. of Bolts | 4 | 4 | 5 | 6 | 6 | 6 | 6 |

Flanged couplings for heavy duty are usually of steel and of more robust design than those shown opposite.

## EXERCISES

(1) Draw, full size, the following views of the assembled muff coupling shown in fig. 1: elevation, half in section on the shaft axis; end view, half in section through a coupling bolt.

(2) Draw, full size, the following views of a C.I. flanged coupling for 2" dia. shafts, showing the flanges bolted together: elevation, upper half in section through a bolt hole; end view looking on the nuts. Show standard keys. Dimension the views.

(3) Make a working drawing of a muff coupling suitable for shafts $4\frac{1}{2}$" dia., as in fig. 2. Show three pairs of bolts with 1" webs between them.

(4) Make a working drawing of a flanged coupling suitable for shafts $3\frac{1}{2}$" dia.

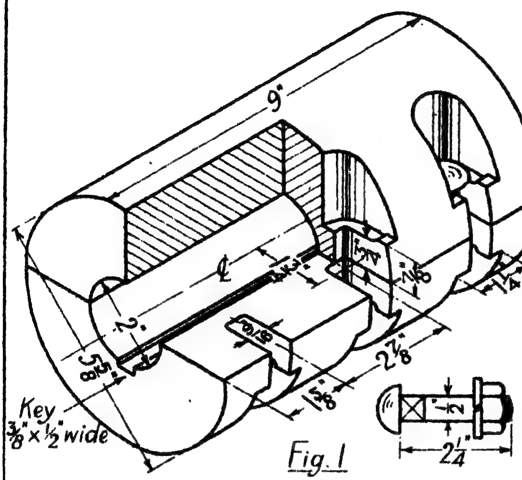



Fig. 1

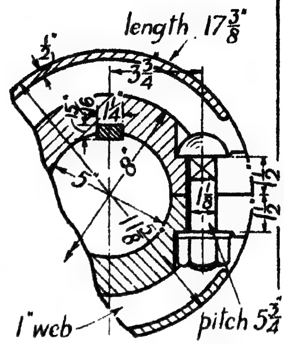



Fig. 2.
Hollow type
for larger shafts.

**MUFF COUPLING**

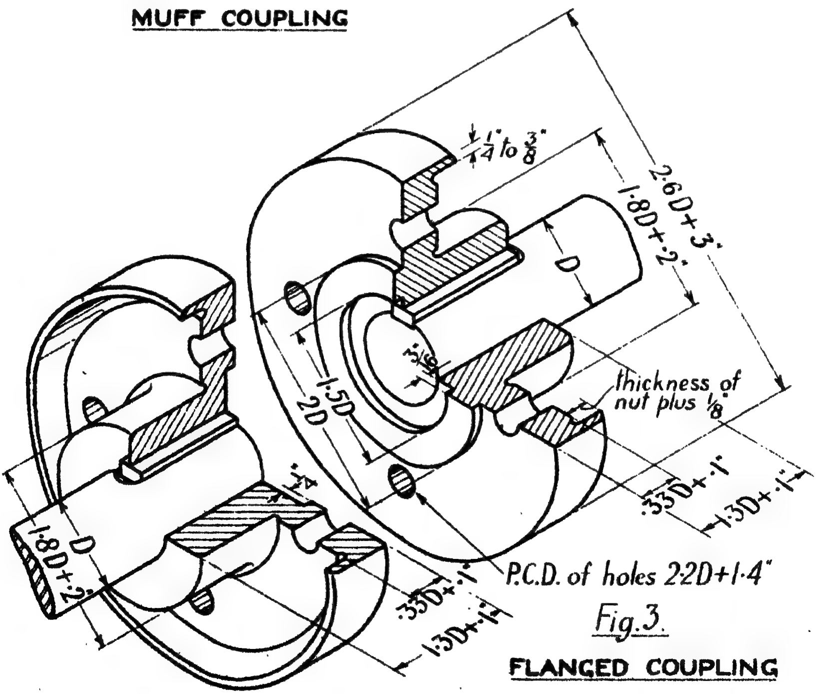



Fig. 3.

**FLANGED COUPLING**

**Claw Couplings or Clutch. Fig. 1.**—This coupling is designed for disengagement at will. The fast half of the coupling is pressed on and keyed to one shaft (B); the other half is arranged to slide along a feather on the other shaft (A), the two halves engaging by means of projecting jaws on the faces. The sliding half is operated by a lever, the forked end of which fits the groove around this part of the coupling.

Shaft (A) enters the first half for a short distance, and is supported by it. When the sliding half is disengaged, shaft (A) remains stationary while the bush of the fast half revolves around it. The ends of the shafts are arranged to be $\frac{1}{8}''$ apart.

Dimensions of a coupling suitable for shafts 2″ dia. are given in fig. 1. For diameters of 3″ upwards two feathers, opposite to each other, are provided for the sliding half. A lubricator of the Stauffer type (see p. 100) is usually fitted to supply grease to the bush.

**Compression Coupling. Figs. 2 and 3.**—This coupling is an alternative to the ordinary flanged coupling. It requires no keys and is easily fitted or removed. The ends of the shafts enter, for equal lengths, a steel sleeve machined externally to a double-conical form. The sleeve is cut through at six equidistant points on the circumference, one saw cut running from end to end, and the others alternately from either end to within 1″ of the opposite end. This sleeve is encased by cast-iron flange-type couplings, fig. 3, bored to the taper on the sleeve: when the halves are drawn together by the coupling bolts, the split sleeve grips the shaft tightly. Power is thus transmitted from one shaft to the other by friction only.

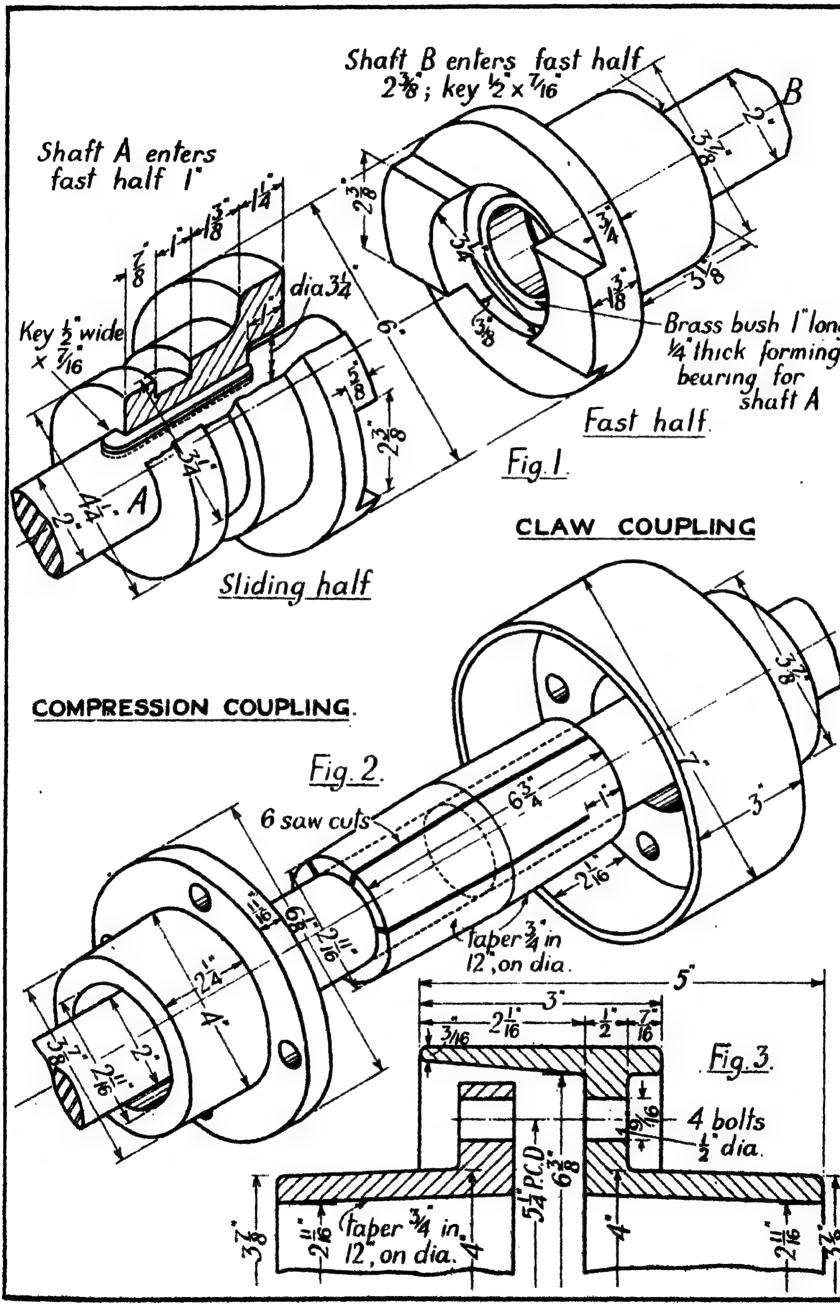
**Loose Coupling for Propeller Shafting.**—This is illustrated on p. 175. The outboard shaft of a ship has to be entered or removed through a tube of small diameter at the stern. The first inboard coupling must therefore be removable or, as it is called, "loose". When the vessel is going astern there is an axial thrust tending to withdraw the shaft from its coupling, and to prevent this, a keep ring in halves is fitted at the end of the shaft.

**Flexible Couplings.**—These are of various types, and are designed to permit slight relative motion between co-axial shafts or to connect shafts which may become slightly out of line. In the coupling shown on p. 174, power is transmitted through driving-pins which carry leather rings, and a small relative axial movement is permissible. This type of coupling is often fitted between a prime mover and a dynamo. In other designs the flanges are connected by a system of leather links; or the bolts are replaced by bundles of thin steel plates.

## EXERCISES

(1) Draw, full size, the following views of the claw coupling in fig. 1, showing the halves engaged: elevation, upper half in section on the shaft axis; end views on the face of each half. Insert all necessary dimensions. The location of the position of the feather key is left to the student.

(2) Draw, full size, the following views of the compression coupling shown in figs. 2 and 3: elevation, upper half in section on the shaft axis; end view from left to right. Show the coupling bolted together and the sleeve projecting $\frac{1}{16}''$ through each flange. Insert dimensions and mark surfaces requiring machining.


Shaft B enters fast half 2 3/8"; key 1/2" x 7/16"
B
Shaft A enters fast half 1"
Key 1/2" wide x 7/16"
dia 3 1/4"
A
Brass bush 1" long 1/4" thick forming bearing for shaft A
Fast half.
Sliding half
Fig. 1.
CLAW COUPLING
COMPRESSION COUPLING.
Fig. 2.
6 saw cuts
taper 3/4" in 12", on dia.
Fig. 3.
4 bolts 1/2" dia.
5 1/4" P.C.D.
Taper 3/4" in. 12", on dia.

**Universal Coupling or Hooke's Joint.**—This coupling is used to connect two shafts whose axes intersect at a small angle, usually less than 30°. It has a wide application in engineering, being used for machine tools, motor-cars, agricultural machinery, &c.

The coupling has many forms. In fig. 1 the shafts are keyed into similar forked ends, which engage with pins arranged on opposite sides of a central cross piece. As shown in fig. 2, one pin passes right through the cross piece and the projections at each end form the "journals" for one forked end. The short pins are arranged at right angles to the long pin; they are driven in, and held either by set screws, bearing on recessed portions, or by tapered pins passing through cross piece and pin. The long pin may be locked in position in the same way, but in the smaller sizes it is assumed to be sufficiently secure without pinning.

**Alternative Designs.**—Refer to fig. 3. The arrangement shown is an improvement on that given in fig. 2. Four similar pins are screwed into the forked ends and locked in position by means of thin check nuts. Plain ends project into the cross piece and form trunnion bearings. Thus arranged, the pins are easily taken out for inspection or on dismantling.

Refer to p. 176. In the coupling shown there, the trunnion pins are cast or forged on the forked ends and fit into bearings arranged in a split enveloping ring.

**Proportions.**—These vary considerably in practice and depend largely on the form of the coupling. The dimensions given in figs. 1 and 2 are suitable for a light-duty coupling for shafts 1¼" dia. Essential dimensions (A, B, C, E, F, G, H, J) are also given in terms of the shaft diameter D. For better-class couplings, G.M. bushes should be provided for the pin bearings and Stauffer type lubricators fitted at convenient positions. Supporting bearings must always be provided as close as possible to each side of the coupling, and the shafts fitted with thrust collars. Frictional losses are low if the coupling is well designed and the working angle small.

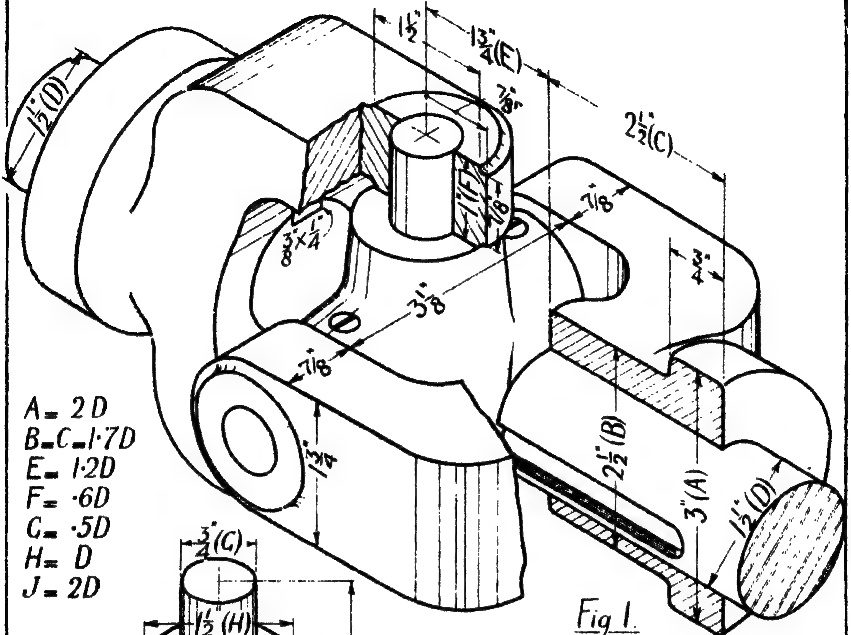
**Double Hooke's Joint.**—The velocity ratio between the shafts is not uniform,* and may cause vibration when the coupling is transmitting power. The variation in velocity ratio may be eliminated by using two Hooke's joints, and driving through an intermediate connecting-shaft. This connecting-shaft must be equally inclined to the two shafts and its own forks must lie in the same plane. The arrangement is known as a double Hooke's joint.

## EXERCISES

(1) Draw, full size, the following views of the coupling shown in fig. 1: sectional elevation, taken through the shaft axes; plan; end view, one half in section through the journals. Use your own judgment for proportions, radii, &c., not given. Dimension the views.

(2) Prepare a working drawing of a universal coupling similar to that shown in fig. 1, but with pins arranged as in fig. 3, to connect two 2½" dia. shafts. Do not provide for bushed bearings and omit lubricators.

* Refer to text-books on "Mechanism" for an analytical discussion.




A= 2D
B=C=1·7D
E= 1·2D
F= ·6D
G= ·5D
H= D
J= 2D

Fig 1.

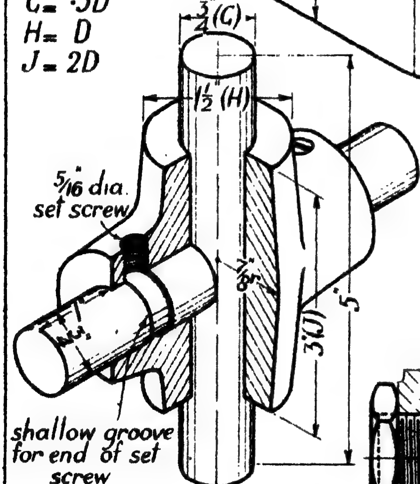



Fig. 2.

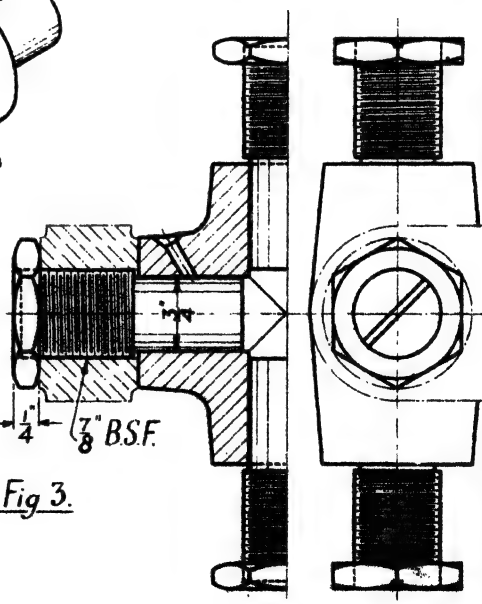



Fig 3.

**Pulleys** are usually made either wholly of cast iron or with a cast-iron boss and a steel rim.* They may be in one piece, or in halves to facilitate assembly on the shaft. The rim may be secured to the central boss either by a web, as in fig. 1, or by radial arms, as in figs. 2 and 3.

The rim of the pulley is frequently "cambered", i.e. the diameter of the pulley is reduced at the edges: this has the effect of keeping the belt in its place on the pulley.† The reduction in diameter at the edges of a pulley 12″ wide is commonly $\frac{3}{8}$″, and pro rata for smaller widths.

The stress in the pulley rim due to centrifugal force is given by $12mv^2/g$ lb./in.$^2$, where $m$ is the mass of 1 in.$^3$ of the material (= 0·29 lb. for C.I.), $v$ is the rim velocity in ft./sec., and $g$ is 32·2 ft./sec.$^2$. A rim velocity of 100 ft./sec. gives a stress of 1080 lb./in.$^2$, and to this must be added the stress caused by the pull of the belt and any stress occasioned by shrinkage in the casting (an important factor in C.I. pulleys). This velocity of 100 ft./sec. is taken as a maximum for all types of balanced pulleys: for *unbalanced* C.I. pulleys 50 ft./sec. is commonly regarded as a maximum.

**Cast-iron Solid Pulleys.**—Usually the diameters $d$, of the shaft, and D, of the pulley, are fixed; other proportions depend upon these dimensions. If D is less than 7″, a solid web is used, as in fig. 1. For diameters from 7″ to 24″, 4 arms may be used; from 24″ to 60″, 6 arms; above 60″, 8 arms. On the assumption that the driving force transmitted by the belt {$(T_2 - T_1)$, refer to fig. 6, p. 69} is in turn transmitted to the boss by all $n$ arms, each arm may be treated as a cantilever of elliptical section carrying an end load of $(T_2 - T_1) \div n$, and its proportions obtained as on p. 84; it is sometimes assumed, however, that the power is transmitted by only half the total number of arms at any given time.

For C.I. pulleys, the outside of the boss and the inside of the rim are slightly tapered ($\frac{1}{8}$″ per foot) to facilitate the removal of the pattern from the mould.

**Split Pulleys** of the type shown in fig. 3 have largely superseded other kinds. The arms are of steel and are shrunk into the hub or boss; the ends pass through the steel rim and are riveted over. The halves of the rim are secured by butt straps, riveted to one half and bolted to the other alternately. The rim is machined after riveting and the pulley finally balanced.

## EXERCISES

(1) Draw, full size, a sectional elevation and end view (i.e. a view along the axis) of a C.I. pulley with a flat rim, as in fig. 1, taking the following dimensions: D = 7″, $d$ = 1″, $b = 2\frac{1}{2}$″, $b_1$ = 2″, $t = \frac{5}{16}$″, $t_1 = \frac{3}{16}$″. Offset the boss to bring one end in line with an edge of the rim and show a keyway. Dimension the views and insert finish marks.

(2) Draw, half size, a sectional elevation and end view of a cambered C.I. pulley, as in fig. 2, taking the following dimensions: D = 18″, $d$ = 2″, $b$ = 6″, $b_1$ = 4″, $h$ = 2″, $t = \frac{1}{2}$″, $t_1 = \frac{3}{8}$″. Dimension the views.

(3) Draw, half size, a view in the direction A of the half pulley shown in fig. 3, showing a part in section on the centre line, and project a plan. Dimension the views.

* Pulleys of pressed steel, wood, and compressed paper are also used.

† A stiff belt running on a coned pulley takes a "set" before contact, so that the portion of the belt about to touch is slightly displaced towards the larger diameter. Hence the belt tends to "creep" along the cone to the larger end.

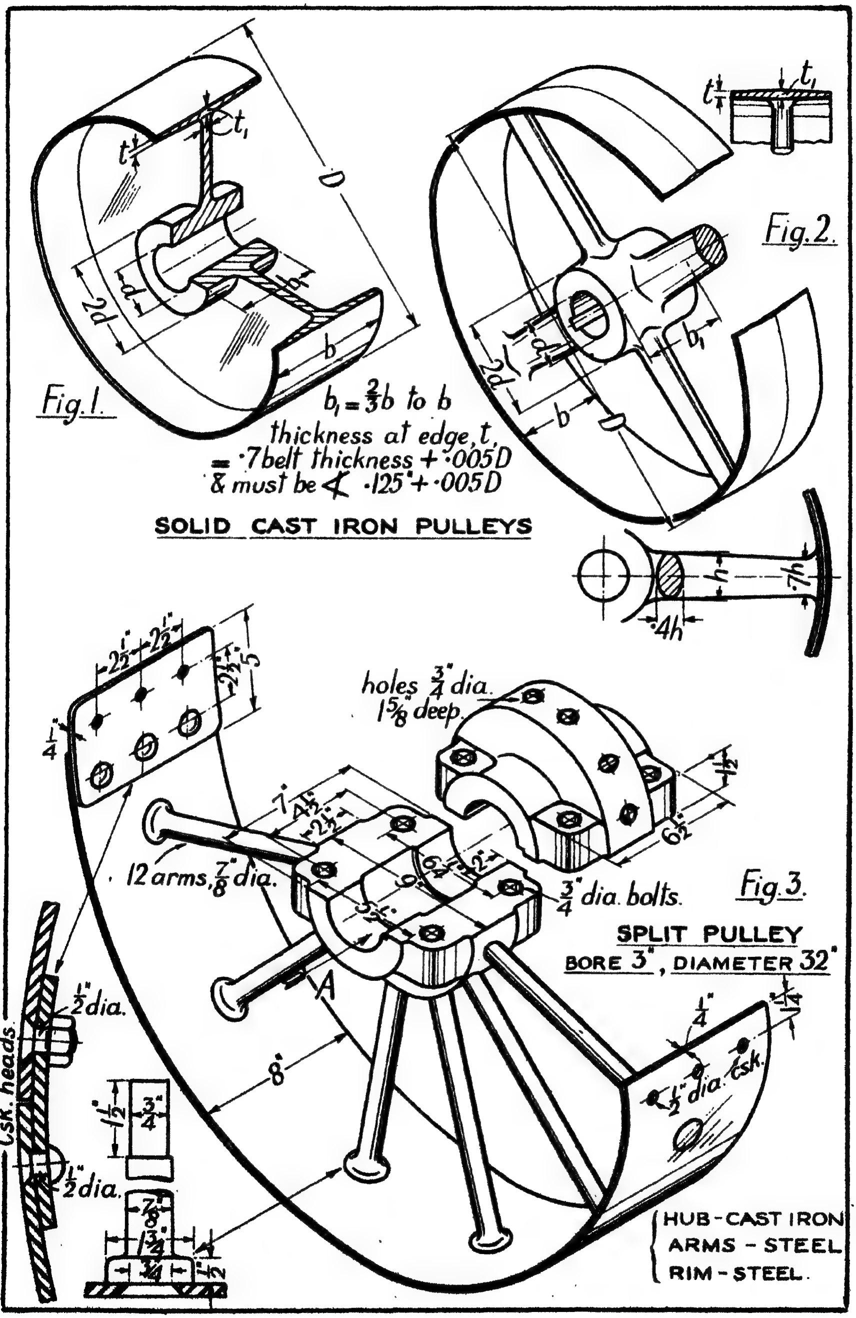


Fig. 1.
Fig. 2.
$b_1 = \frac{2}{3}b$ to $b$
thickness at edge, $t$,
$= \cdot 7$ belt thickness $+ \cdot 005D$
& must be $\nless \cdot 125'' + \cdot 005D$
SOLID CAST IRON PULLEYS
holes $\frac{3}{4}''$ dia. $1\frac{5}{8}''$ deep.
12 arms, $\frac{7}{8}''$ dia.
$\frac{3}{4}''$ dia. bolts.
Fig. 3.
SPLIT PULLEY
BORE 3", DIAMETER 32"
Csk. heads.
$\frac{1}{2}''$ dia.
$\frac{1}{2}''$ dia. csk.
HUB - CAST IRON
ARMS - STEEL
RIM - STEEL.

**Fast and Loose Pulleys,** fig. 1, are mounted on a driven shaft when it is desired to stop and start the shaft without interfering with the driving shaft. The fast pulley is keyed, but the loose pulley revolves idly on the shaft. To stop the shaft, the belt is shifted from the fast to the loose pulley by means of "striking gear", a fork engaging with the belt on the leading side just before it reaches the pulley. The diameter of the loose pulley is sometimes made slightly less than that of the fast pulley to relieve the belt of its tension. To enable the belt to mount easily the larger pulley, one rim is bevelled at the edge, either as shown in fig. 2 or as in fig. 3.

In fig. 1 end movement of the pulleys in one direction is prevented by a steel collar which is secured to the shaft by a grub screw; the face A bears against the supporting bracket and prevents movement in the opposite direction.

**Stepped Pulleys, or Speed Cones,*** fig. 4, are fitted to enable the speed ratios of two shafts to be varied; refer to fig. 1, p. 69. By shifting the belt from one pair of pulleys to another, the speed ratio of the shafts is changed. The same belt is to be used for all pairs; hence the diameters of the steps must be such that no appreciable change in length is occasioned. This condition is satisfied, exactly for crossed belts and very nearly† for open belts, if the sum of the radii of corresponding pulleys is constant. If the speed ratios be denoted by $S_1$, $S_2$, $S_3 \ldots$, the radii of the driving pulleys by $R_1$, $R_2$, $R_3$, . . ., and the radii of the driven pulleys by $r_1$, $r_2$, $r_3$, . . ., then the following equations must be satisfied:

$$S_1 = R_1/r_1,\ S_2 = R_2/r_2,\ S_3 = R_3/r_3, \text{ \&c.,}$$

and

$$(R_1 + r_1) = (R_2 + r_2) = (R_3 + r_3), \text{ \&c.}$$

**Horse-power transmitted by Belting.**—If $T_2$ and $T_1$ are the total tensions in the tight and slack slides of a driving belt, the horse-power transmitted is given by horse-power $= \{(T_2 - T_1)\pi DN\} \div (33{,}000 \times 12)$; or horse-power $= (T_2 - T_1)V \cdot 60 \div 33{,}000$; where $T_2$ and $T_1$ are in pounds, D is the pulley diameter in inches, N the shaft speed in revs./min., and V the belt velocity in ft./sec.

The ratio $T_2/T_1$ lies between 2 and 3. For ordinary belting a safe maximum stress is 300 lb./in.$^2$. Belt speeds in machine shops average about 20 ft./sec. At speeds above 40 ft./sec. the additional tension produced by centrifugal force must be taken into account.

## EXERCISES

(1) Draw, half size, an elevation, half in section, of the pulleys in fig. 1. Keyed to the same shaft show a speed cone similar to that in fig. 4 (but suitable for the smaller shaft); arrange the cone with its large end 6″ from the side of the loose pulley and show it half in section. Dimension the drawing.

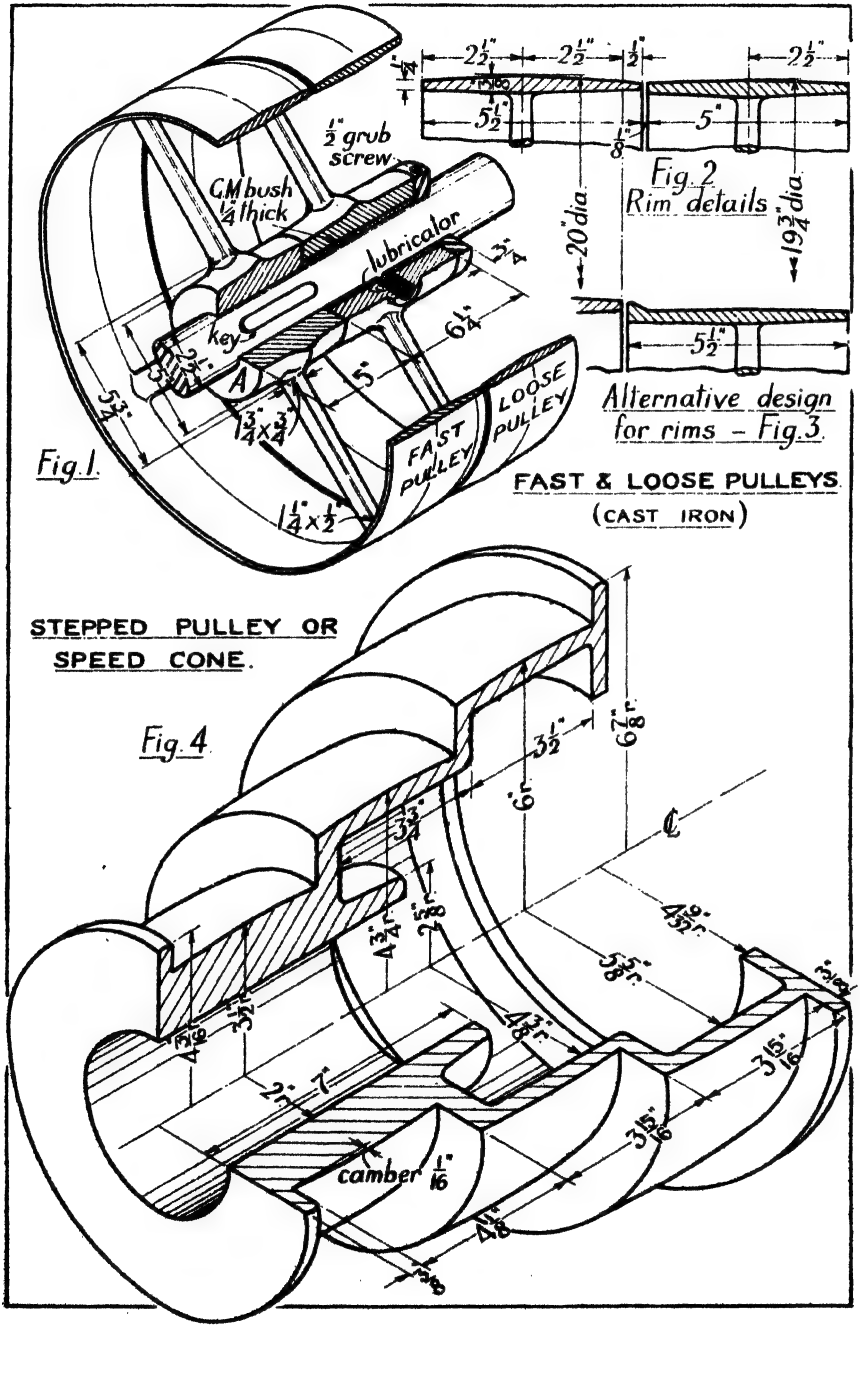
(2) If the pulleys are driven by a belt $3\frac{1}{2}'' \times \frac{1}{4}''$ and the stresses in the tight and slack sides are 205 and 80 lb./in.$^2$, calculate the horse-power transmitted at a velocity of 20 ft./sec.

*Answer.*—4 *h.p.*

* Geared headstocks have largely replaced stepped pulleys on machine tools. The pulleys shown are suitable for a 6″ lathe.

† The error is negligible if the distance $d$ between the shafts is large compared with the difference between corresponding pulley diameters. The length of belt $l$ is given approximately by

$$l = 2d + \pi(R_1 + r_1) + \{(R_1 - r_1)^2/d\}.$$




FAST & LOOSE PULLEYS (CAST IRON)

STEPPED PULLEY OR SPEED CONE.

**Wire Ropes** are extensively used in lifting and transporting appliances, and their length of service depends largely upon the design of the pulleys around which they pass. Much research work on wire ropes has been carried out by the Institution of Mechanical Engineers,* and the Reports of the Research Committee show that for long service ***the pulley diameter must be as large as possible.***

A small increase of pulley diameter has a beneficial effect on the rope; indeed, it may be taken that the addition of 2 rope circumferences to the diameter of the pulley will double the life of the rope.

From the Reports it appears that the correct pulley diameter is more closely related to the diameter of the rope than to the diameter of the wire of which the rope is made. Many-stranded ropes are superior to others of the same diameter with fewer strands, only when the working conditions are severe.

A generally accepted rule in crane practice is that the diameter of the smallest pulley should not be less than 22 rope diameters,† and with a standard type of rope this gives a pulley diameter/wire diameter ratio of 330/1. In the experiments referred to, pulleys of less than 300 wire diameters were found to misuse the rope. It would appear preferable, therefore, to take a higher ratio, say 500/1, as the ***minimum.***

The **groove** in the pulley should be machined to a gauge having a radius equal to, or not less than, that of the rope, and "flared" on tangents to the rope circle, as in fig. 1. If the rope does not bear on the bottom of the groove, lateral crushing occurs and the rope is injured.

The C.I. pulley shown, fig. 2, has an oil reservoir incorporated in the body, three filling plugs being provided. The proportions of the casting are entirely empirical and represent good practice.

**Telodynamic Transmission** is the transmission of power by wire ropes. A typical pulley-rim section is shown in fig. 3; the bottom of the groove is lined with leather (or with wood or gutta-percha). The ratio pulley diameter/wire diameter is large, frequently 2500/1. The wire velocity varies from 60 to 100 ft./sec.

**Stresses in Wire Ropes.**—The total stress $f_m$ in a wire rope is the sum of the direct tensile stress $f_t$, due to the load, and the stress due to bending over the pulleys, $f_b$; the latter is never negligible, and may exceed the former if the pulley diameter is small.

Hence $\qquad f_m = f_t + f_b$,

i.e. $\qquad f_m = T/A + c \,.\, E \,.\, d/D$,

where T is the total rope tension in lb., A is the total sectional area of the wires in in.$^2$, E is Young's modulus in lb./in.$^2$, $d/D$ is the ratio largest wire diameter/pulley diameter, and $c$ is a constant. E is about 30,000,000 lb./in.$^2$, and $c$ varies from 0·35 to 1. The stress $f_m$ should not exceed the fatigue limit of the wire. This limit depends to a great extent on the rope arrangement, and if reverse bends are present its value may not exceed 38,000 lb./in.$^2$. Reverse bends should be avoided, as they halve the life of a rope: e.g. the reverse bend arrangement in fig. 4 may be improved as in fig. 5.

## EXERCISE

Draw, half size, views in the directions A and B of the half crane pulley shown and dimension them. If the rope laps 180° of the pulley and sustains a tension of 360 lb., what is the average pressure on the bush? If the rope has 6 strands, each of 6 wires of dia. 0·036″, calculate $f_m$, taking the constant $c = 0{\cdot}35$.

*Answer.*—$p = 80$ *lb./in.*$^2$; $f_m = 35{,}200$ *lb./in.*$^2$.

* *Proc. I. Mech. E.*, 1920, 1924, 1928, 1930, 1935 (Final Report).

† On the Manchester Ship Canal Co.'s high-speed cranes the ratio is 35.

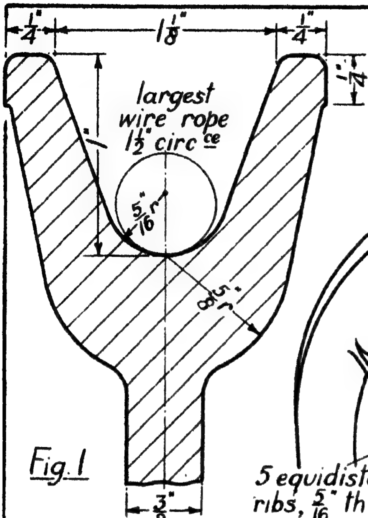
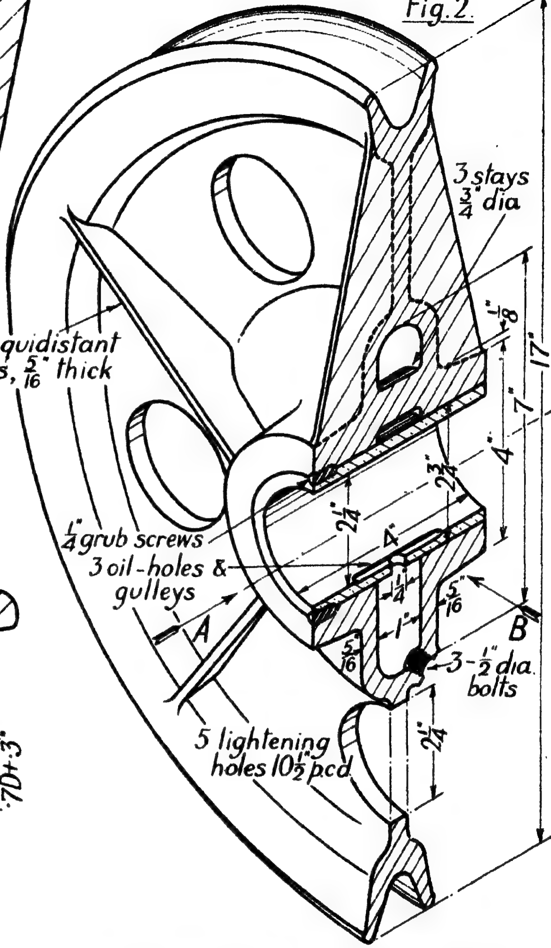



Fig. 1

SECTION OF RIM
FULL SIZE (APPROX)

CRANE PULLEY
RESERVOIR TYPE

Fig. 2



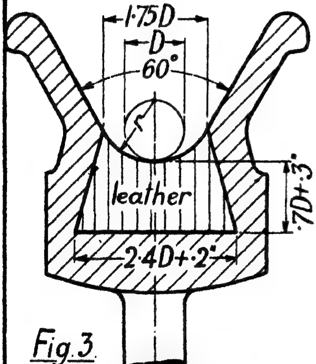




Fig. 3

PULLEY RIM
TELODYNAMIC
TRANSMISSION TYPE

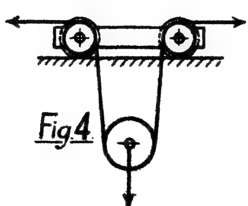


Fig. 4

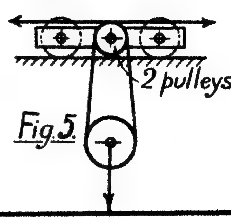



Fig. 5

**Linkages.**—This name covers the various mechanisms which are employed to convert rotary motion into oscillating or reciprocating motion. They are usually combinations of levers, connecting rods, and cranks, and are applied to machine tools, pumps, steam-engines, &c. The design of some of these parts is discussed here. In the **four-bar chain** or linkage, fig. 1, a crank A rotates about O and, by means of the connecting rod B, causes an arm C to oscillate about Q. In the **slider-crank chain,** fig. 2, a crank A rotates about O and causes a crosshead C to reciprocate between guides.

**Proportions of Simple Levers and Cranks.**—In what follows all dimensions are in inches, loads (P) in lb., and stresses ($f$) in lb./in.$^2$.

**Simple Lever. Fig. 3.**—A force P, applied at the end of the lever, produces on the shaft S a twisting moment, magnitude P . R, together with a bending moment, magnitude P . N, where N is the overhang of the centre of the lever to the centre of the nearest bearing. (If the lever is supported between bearings, the magnitude of the bending moment is given by the product of the reaction at a bearing and the distance of the lever from that bearing.) The diameter D of the shaft is settled by applying the formulæ on page 68; the proportions of the boss B, in terms of D, are given in the figure.

The section of the arm at G must be designed for a bending moment of magnitude P . L. For light levers the arm may be elliptical in section, but for heavily loaded levers the I form, fig. 4, is preferable. The moment of resistance of the elliptical section is given by $\pi . b . h^2 . f/32$, where $h$ and $b$ are the lengths of the major and minor axes of the ellipse; usually $b = 0{\cdot}4h$, so that the moment of resistance is $\pi . h^3 . f/80$, and $h$ is given by the formula

$$P . L = \pi . h^3 . f/80. \quad . \quad . \quad (1)$$

The dimensions at the small end depend largely upon the manner in which the load P is applied and definite rules cannot be given. Suitable proportions in terms of the diameter $d$ of the pin at the small end are shown in the figure.

**Bell Crank. Fig. 4.**—This is a combination of two cranks, usually at right angles. The proportions of the boss B depend on the diameter of the shaft. When proportioning the arms it is usual to assume that the bending moment is taken by the flanges only, i.e. the effect of the web is neglected. On this assumption the moment of resistance to bending is given by (flange area × depth × stress): hence

$$P . L = b . t . h . f. \quad . \quad . \quad (2)$$

## EXERCISES

(1) Draw, full size, an elevation of the lever in fig. 3, showing a turned section, and an end view. Dimension the drawing. Take D = 2", $d$ = 1¼", R = 8", $h$ = 3", and least depth of lever = 1⅞". If P = 350 lb. and N = 4", calculate the greatest stresses in the arm and shaft.

*Answer.—2000 and 4000 lb./in.$^2$.*

(2) Prepare, full size, a working drawing of the bell crank (fig. 4). Show each arm broken to reveal the section and shortened to bring the drawing within the limits of the paper. If $f$ is not to exceed 2000 lb./in.$^2$, calculate the safe load which may be applied to the end of an arm.

*Answer.—389 lb.*

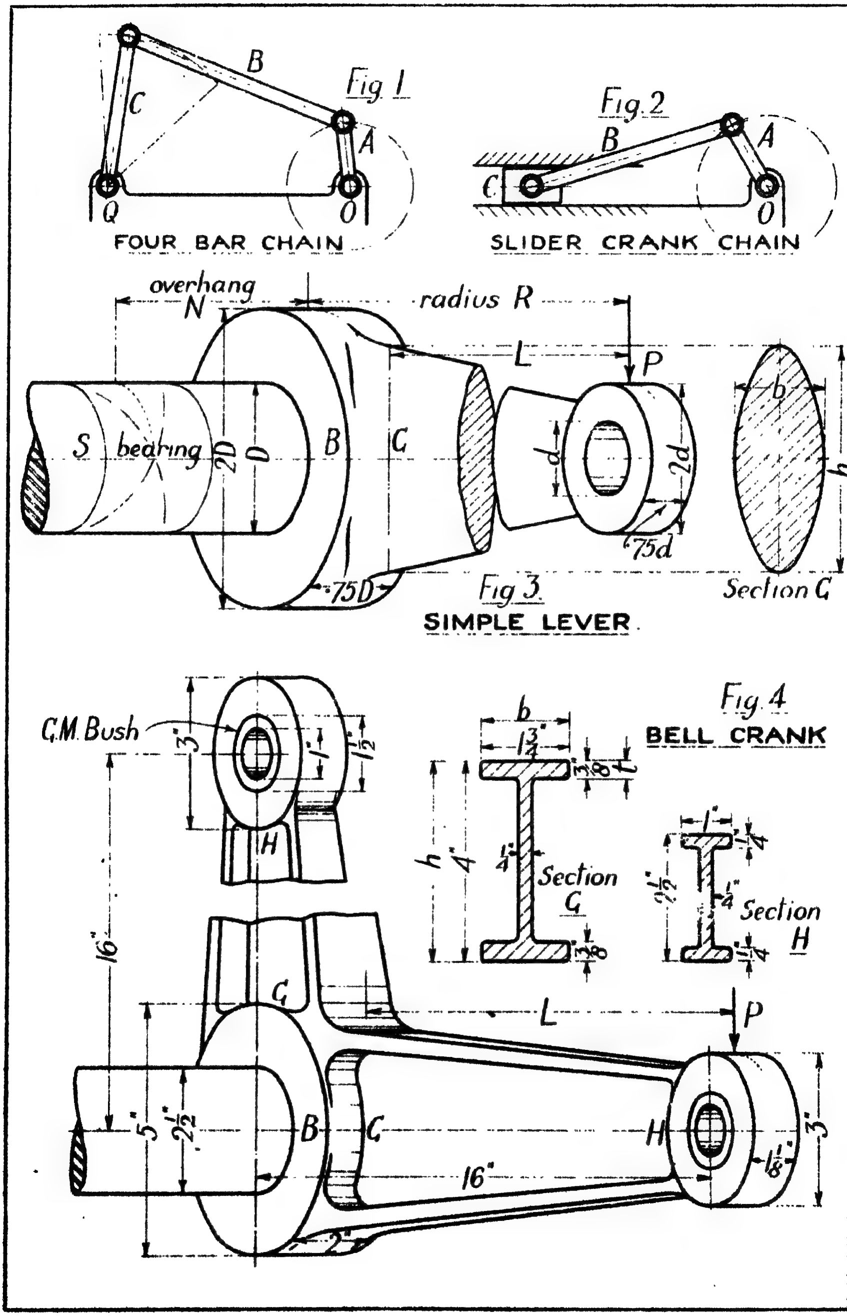


Fig. 1
FOUR BAR CHAIN
Fig. 2
SLIDER CRANK CHAIN
overhang N
radius R
bearing
·75D
·75d
Section C
Fig. 3.
SIMPLE LEVER.
G.M. Bush
Fig. 4
BELL CRANK
Section G
Section H

**Overhung Crank. Fig. 1.**—In the design shown, all parts are of steel. The crank arm is either forced on the shaft under hydraulic pressure, or heated and shrunk on, and then keyed. The crank pin is pressed into the eye and riveted over. In a light-duty crank the arm may be of cast iron, but it should then be webbed to give a trough-shaped section.

**Proportions.**—The proportions of the **bosses** are fixed by the diameters $d$ of the pin and D of the shaft: they are given in terms of these dimensions. If the **crank pin** is proportioned to give a sufficient bearing surface, it will usually have an excess of strength. The length $l = 0{\cdot}8d$ to $1{\cdot}3d$. Taking the latter value, the projected area $= 1{\cdot}3d^2$.

Hence

the total load P =

$$1{\cdot}3d^2 \times \text{safe bearing pressure.}$$

Bearing pressures for cranks of this type vary considerably in practice according to the class of work and the speed. An average range is from 600 to 1000 lb./in.$^2$.

Having obtained $d$, the pin must be checked for strength. The bending moment on the pin $= \frac{1}{2}\mathrm{P}\,.\,l$*; in the equation

$$\tfrac{1}{2}\mathrm{P}\,.\,l = \pi\,.\,d^3\,.f/32,$$

$f$ should not exceed 8000 lb./in.$^2$.

The **shaft** is subjected to combined bending and torsion, and its diameter may be calculated as on page 68 for given maximum values of M and T. Frequently, however, the diameter is based on the H.P. to be transmitted. From the H.P. the ***mean value*** of the twisting moment T is given by the equation

$$\text{H.P.} = \mathrm{T} \times 2\pi\,.\,\mathrm{N} \div (33{,}000 \times 12)\ldots \quad \text{Eqn. 5, page 68} \quad . \quad . \quad (1)$$

In this, $\mathrm{T} = \pi \mathrm{D}_1^3\,.f/16$, where $\mathrm{D}_1$ is the nominal shaft diameter; $f$ may be taken as 8000. Substituting in (1),

$$\text{H.P.} = (8000\,.\,\pi\,.\,\mathrm{D}_1^3)(2\pi\,.\,\mathrm{N}) \div (16\,.\,33{,}000\,.\,12),$$

from which

$$\mathrm{D}_1 = 3{\cdot}42\sqrt[3]{\text{H.P.} \div \mathrm{N}}\,.\quad . \quad . \quad (2)$$

The greatest twisting moment is much larger than the mean, in some engines being twice as great. To allow for this variation and also for bending moments, the diameter $\mathrm{D}_1$ in equation (2) is multiplied by a factor which varies from 1·34 to 1·49. Taking the lower value, the diameter required is given by

$$\mathrm{D} = 4{\cdot}58\sqrt[3]{\text{H.P.} \div \mathrm{N}}\,.\quad . \quad . \quad (3)$$

The **crank arm** is subjected to combined bending and torsion and for the shape considered the stress analysis is complex. An approximation is given by treating the arm as a column under a load P and a bending moment P.N, this being the dead centre loading.

**Disc Cranks. Fig. 2.**—This type is commonly used on small engines and pumps. The disc is of cast iron and a surplus weight is provided opposite the crank pin for balance.

## EXERCISES

(1) Draw, half size, two views corresponding to those in fig. 1 of a steel overhung crank, taking $d = 3''$, $\mathrm{D} = 4\frac{1}{2}''$, $\mathrm{R} = 12''$, $l = 1{\cdot}3d$. Show a key $1\frac{1}{2}''$ dia. (B.S.Fine). If P = 7000 lb., calculate the bearing pressure and the greatest stress in the pin. State the maximum horse-power for which the shaft should be used, at 200 revs./min.

*Answer.*—600 *lb./in.*$^2$ *approx.*; 6737 *lb./in.*$^2$ (*at inner root*); 200 *h.p. approx.*

(2) Draw, half size, the following views of the disc crank (fig. 2): elevation, on back of disc; sectional side elevation through the axes; plan, half in section through the shaft axis. Dimension the views and insert finish marks. If the greatest stress in the pin is not to exceed 7000 lb./in.$^2$, calculate the greatest permissible load and state the bearing pressure.

*Answer.*—1785 *lb.*; 595 *lb./in.*$^2$.

* For the design shown it would be safer to take the bending moment as $\mathrm{P}(\cdot 5l + \cdot 1d + \cdot 3'')$.

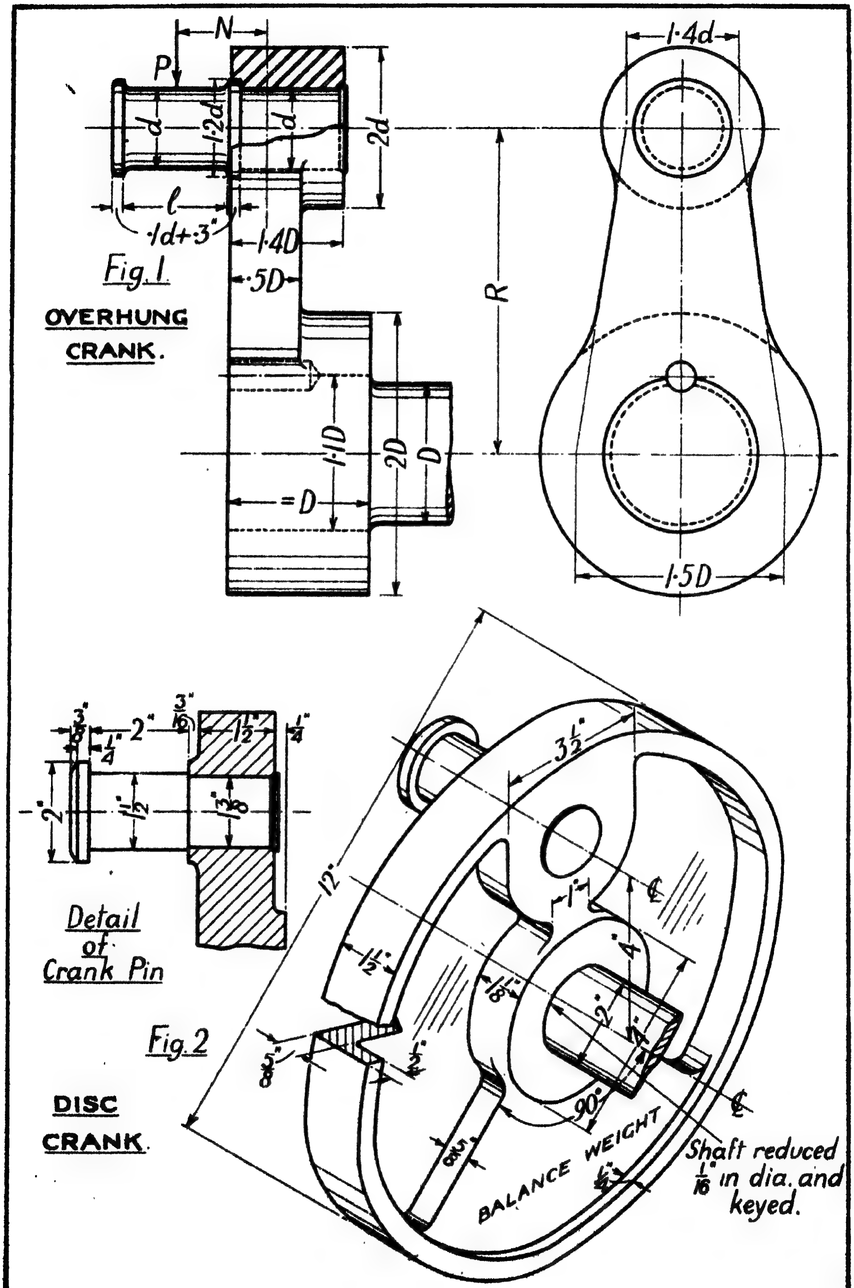



Fig. 1

OVERHUNG CRANK.

Detail of Crank Pin

Fig. 2

DISC CRANK.

**Marine Type Cranks. Fig. 1.**—The drawing gives proportions for steel cranks, forged in one piece, as used in marine and internal-combustion engines. There are usually more cranks than one, and a complete shaft may be either in one piece or in parts coupled together. Two cranks are usually arranged at right angles to each other, and three cranks at 120° to one another. The diameter of the shaft is given by

$$D = c\sqrt[3]{H.P. \div N},$$

where $c$ varies from 4·5 for 3 cranks to 5·1 for 1 crank. The proportions of the crank arms are settled by empirical formulæ based on successful practice, the stresses in the arms being too complex for exact treatment.

The design shown in fig. 1 is not satisfactory for very large engines, the cranks of which are invariably built up. The proportions of large marine crank shafts are usually based upon rules laid down by the Registry Associations. A typical design is given on page 173, the arms being shrunk on to the shafts and crank pins, and keyed: usually the shafts and pins are hollow.

An interesting modern design of crank pin is given on page 157.

**Locomotive Crank. Fig. 2.**—Details are given of a balanced locomotive crank shaft of modern design, all parts being of steel. The webs are shrunk on and keyed: for the shaft shown the allowance for shrinkage = (pin diameter) ÷ 800.

The pin and arms of a crank of the type shown in fig. 1 set up disturbing forces as they revolve at high speed: if the total weight of the displaced parts is W lb., and the distance of their centre of gravity from the shaft axis is R in., a centrifugal force of $W \cdot R \cdot \omega^2/12 \cdot g$ lb. is produced, $\omega$ being the shaft speed in radians/sec.* This effect is neutralized in the design shown in fig. 2 by the provision of balance weights of suitable mass and displacement, arranged to give an equal and opposite centrifugal force.

Actually, not only are the crank pin and arms balanced by revolving masses, but also part of the connecting rod and part of the reciprocating masses in the engine, balance weights additional to those at the crank being provided on the wheels. The question of balancing cannot be discussed fully here, and the student is referred to standard works on the subject.†

**Automobile Crank Shaft.**—A design for a modern automobile engine is given on page 172. The shaft is a steel stamping and is forged from a solid bar of high-grade steel. The crank arms are not machined and the journals and crank pins are ground to size.

## EXERCISES

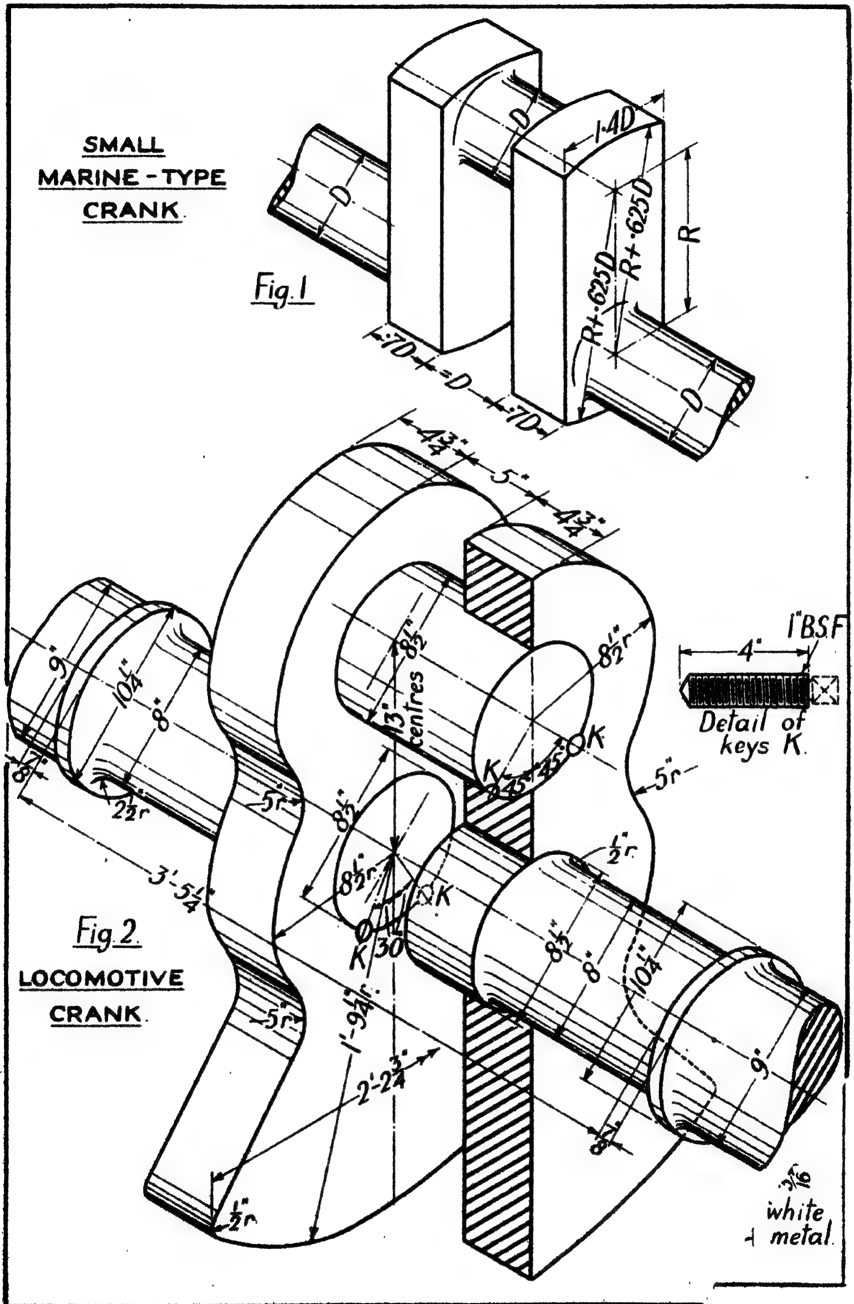
(1) A marine crank shaft, 6″ dia., of the type shown in fig. 1, has two cranks at right angles and is forged in two parts. Each part has an overall length of 38″ and has a coupling, $12\frac{3}{4}''$ dia. × 2″ thick, forged at each end: the crank is centrally arranged. Draw, to a scale of one eighth, an elevation and end view of the complete shaft with the halves bolted together; show 4 coupling bolts, $1\frac{3}{8}''$ dia. (screwed $1\frac{1}{8}''$ dia.) on a P.C.D. of $9\frac{1}{4}''$. Dimension the drawing.

(2) Draw, to a scale of one eighth, an elevation and end view of the locomotive crank shaft shown in fig. 2; in the elevation show one crank arm in section, as in figure. Determine the least temperature difference necessary between the crank pin and arm for the entry of the pin. Take the coefficient of linear expansion for steel ·000006 per degree F.

*Answer.*—209° *F.* (Note.—This temperature difference would be increased in practice to give clearance.)

* Or $WR(2\pi N)^2 \div \{12g \times (60)^2\}$, where N is the speed in revs./min.

† See *Balancing of Engines*, Dalby.




SMALL MARINE-TYPE CRANK.

Fig. 1

Fig. 2.

LOCOMOTIVE CRANK.

**An Eccentric** is a special crank of small radius with a crank pin larger in diameter than the shaft. The **sheave** is virtually the crank pin, and the **strap**, which is in two parts, corresponds to the ordinary rod-end bearing—see pages 141 and 143. Eccentrics are used to convert rotary into reciprocating motion, but not vice versa. They are of various types and are employed to operate valves, pump plungers, &c.

For small eccentrics the sheave is often solid, and of steel. For larger types the sheave is in two parts and is commonly of cast iron. The straps are of gunmetal, cast iron, or of cast iron lined with whitemetal (as discussed later); cast iron is quite satisfactory in practice. A small C.I. strap is shown on page 33.

The distance between the axes of the shaft and sheave is known as the throw, or the radius, or the eccentricity.

**Eccentric for Locomotive.**—Fig. 1 shows a typical design suitable for the shaft on page 89. The sheave is in two parts, the larger of cast iron and the smaller of steel, held together by cottered bolts and keyed to the shaft. The bolts, fig. 3, are a driving fit in the steel part; the cavity beyond the head of each bolt is filled with whitemetal, hammered in. Two set screws with hardened cup-ends are screwed through the larger part and bite into the shaft. The cast-iron strap is flanged to keep it in position on the sheave, and is in halves bolted together. Brass strips are provided for taking up wear. The eccentric rod (not shown) is attached to the strap at A by means of two bolts fitting into recesses in the strap, fig. 2.

The actual design has been slightly simplified for the purpose of providing the exercise below. For example, actually the strap should not be rectangular in section but rounded at the edges: it should also be thickened at the middle. Lubrication details are not fully shown on the drawing. Two lubricating boxes are required, one on each half-strap; also oil pads should be provided, fitting in recesses formed in the lower part of each strap and pressing against the sheave under the action of light springs.

**Proportions.**—The diameter of the sheave, which should be kept as small as possible to reduce the rubbing velocity, depends upon the eccentricity and the least permissible thickness of metal in the sheave: the latter may be kept low when steel is used for the smaller portion (or the whole) of the eccentric. The projected area (diameter × width) of the sheave must be sufficiently large to give an average bearing pressure, due to the load on the strap, of from 80 to 100 lb./in.$^2$. (Bearing pressures are discussed more fully in the section on bearings.) The straps are proportioned from empirical rules to give a stiff design, and the strap bolts should be designed to take the greatest pull in the eccentric rod.

## EXERCISE

di.
cra
scale
view c
bolted
1$\frac{3}{4}$″ dia. ...se the eccentric sheave in fig. 1 to
9$\frac{1}{4}$″. Dim... through an angle of 90°, relative
... Then draw, half size, the fol-
... of the assembled parts: eleva-
...ew in the direction of the
...wing the upper half of sheave and strap in section to reveal the bolts; sectional end view on the shaft axis, from right to left; an end view from left to right. Omit the lubricating box, but show set screws and cotters in position. Dimension the views.

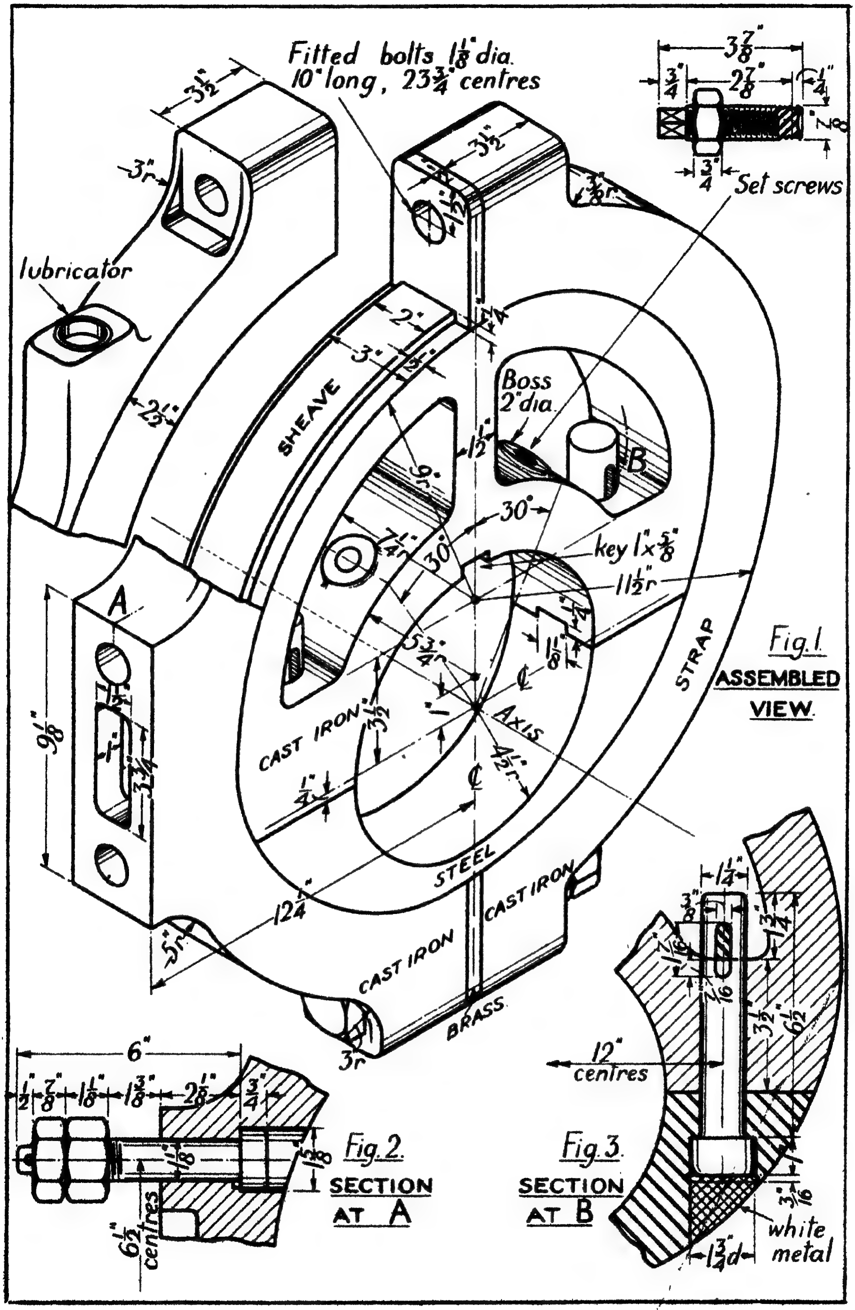



Fig. 1. ASSEMBLED VIEW.

Fig. 2. SECTION AT A

Fig. 3. SECTION AT B

A ***cam*** is a rotating or oscillating machine part designed to communicate reciprocating or intermittent motion, often of a comparatively complicated kind, to another machine part, called a ***follower***. Cams may be divided into two main groups, Disc or Radial Cams, and Cylindrical Cams. Only the former are dealt with here, but the treatment of the latter presents no additional difficulties. Other special types of cam are in use, e.g. conical and spherical cams. Because almost any desired motion can be given to the follower, cam design is of great importance in mechanical engineering.

**Types of Radial Cams.**—The V-ended follower in fig. 1 is not of practical importance, but it gives the ***pitch curve***, from which the working curve for other cams is obtained. The pitch curve may be defined as the line which would give a V-ended follower the desired motion. In practice, roller-ended followers, as shown in figs. 2*a*, *b* and *c*, are commonly used. For these, the working surface usually closely follows the pitch curve. In fig. 2*a* the axis of the follower passes through the cam axis, but in 2*b* it is offset. In fig. 2*c* a groove is used instead of an edge.

The flat-ended or mushroom follower in fig. 3 has its axis offset a little from the plane of the cam in order to reduce friction between surfaces—the contact producing rolling as well as sliding. Fig. 4 shows an oscillating cam with an involute contour—see Ex. 5, p. 96.

All the followers except that in fig. 2*c* will require to be spring loaded to ensure continuous contact.

**Cam Displacement Diagrams.**—Fig. 5 shows a radial cam capable of giving a lift L to a V follower. The intercepts between the profile and the ***base circle*** on the twelve equally spaced radial lines, are set off in fig. 6 as ordinates from a base line of any length representing 360°; this is a ***cam displacement diagram***. The converse operation, of being given, or selecting, the shape of this diagram, and of translating it into a cam profile, is the usual design problem.

The ***way*** in which the follower moves is of great importance in design, and displacement diagrams for three kinds of movement are shown opposite. In each it is assumed that the lift occurs during 180° turning of the cam; that there is then a "rest" or "dwell" period of 60°; and that the follower returns to its starting point during the next 120°. These spacings have been chosen simply to give clear diagrams.

In fig. 6 the diagram has straight inclined lines, so that the velocities during lift and fall are constant (but not equal).

The diagram in fig. 7 will give the follower simple harmonic motion (S.H.M.). For its construction, the semicircle, radius $r = L \div 2$, is divided equally and perpendiculars are dropped on to the diameter. The intercepts on the diameter are the ordinates for the displacement curve, which is a sine curve. The radius of the circle is referred to later as the "***representative crank***"

The diagram in fig. 8 differs little from that of fig. 7 but the difference is important. Fig. 8 gives to each movement of the follower, first uniform acceleration and then uniform retardation. Each curve consists of two parabolas. The projection of points will be clear from the figure on the left, which uses a common method of drawing parabolic curves.

In order to make a choice between these displacement diagrams (and any others) it is necessary to construct corresponding diagrams of velocity and acceleration. From the acceleration diagrams can be deduced the forces which the cam will bring into operation, and which at high speeds are often deciding factors. This is fully discussed on page 94, from which the significance of figs. 6, 7 and 8 will become apparent.

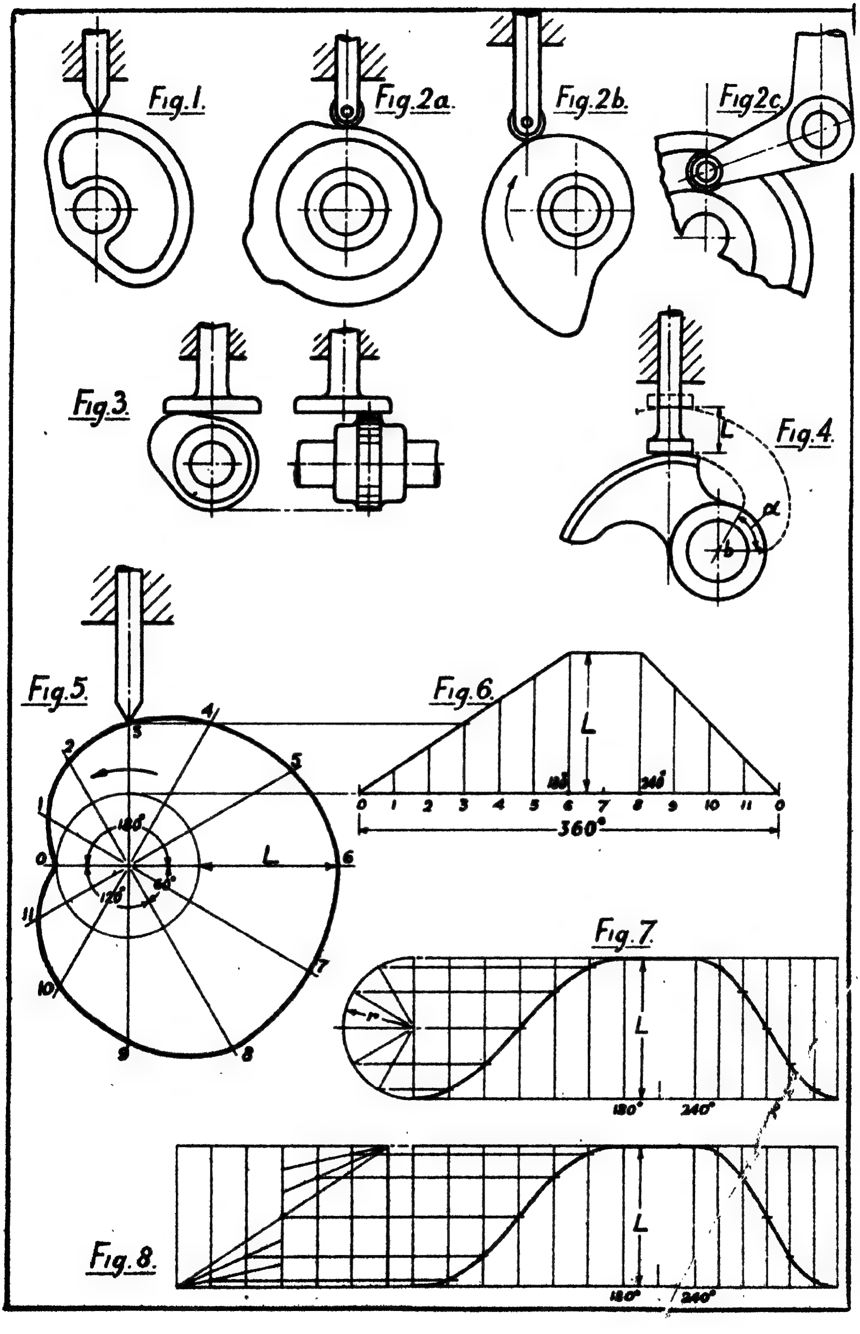


Fig.1.
Fig.2a.
Fig.2b.
Fig2c.
Fig.3.
Fig.4.
L
α
b
Fig.5.
L
0
1
2
3
4
5
6
7
8
9
10
11
180°
120°
60°
Fig.6.
L
180°
240°
0 1 2 3 4 5 6 7 8 9 10 11 0
360°
Fig.7.
r
L
180°
240°
Fig.8.
L
180°
240°

**Displacement, Velocity and Acceleration Diagrams.**—As has already been said, it is necessary to know the maximum acceleration which a cam will give a follower in order that the forces involved may be calculated.

The displacement diagrams of figs. 6, 7 and 8 on the previous page have been reproduced opposite in figs. 1*a*, 1*b* and 1*c* to a smaller horizontal scale. Below them have been drawn corresponding diagrams for velocity and acceleration.† The same scale applies to each set of three. In what follows the type of motion will be referred to by its abbreviation (C.V., S.H.M., or U.A.R.).

The S.H.M. displacement, velocity, and acceleration diagrams are all of the sine curve family. The U.A.R. displacement parabolas give triangular velocity, and rectangular acceleration, diagrams. Students with a knowledge of the calculus will recognize that the fig. 1 curves are the *integral curves* of the corresponding fig. 2 curves, and the fig. 2 curves the integrals of the fig. 3 curves. Conversely, the fig. 2 curves are the *differential curves* of the fig. 1 curves, and the fig. 3 curves the differentials of the fig. 2 curves.†

The velocity diagram for the C.V. cam is rectangular, fig. 2*a*. The acceleration diagram, fig. 3*a*, indicates values infinitely great where the velocity changes in zero time—which is impossible in practice. The displacement curve is often modified as shown dotted, giving corresponding changes in the velocity and acceleration curves.

**The Profiles Compared.**—We are now able to compare the profiles as regards maximum values of (*a*) velocity and acceleration, and (*b*) side thrust on the follower.

*Re* (*a*): The C.V. cam has a lower velocity than the maxima in the other two cases, the *ratios* (not the actual values) being $2:\pi:4$; the abrupt changes however at the ends of the motions would be disturbing at high speeds. S.H.M. gives a lower maximum velocity than U.A.R. ($\pi:4$), but a greater maximum acceleration ($\pi^2:8$). On the other hand, the *total abrupt change* in acceleration is greater for U.A.R. than for S.H.M., the ratio being $(8+8):\pi^2$.

*Re* (*b*): The value of the side thrust on the follower (except when this is flat ended) depends on the slope of the cam profile. The *comparative* values of the slopes are given by the ordinates in the velocity diagrams,† and maximum values are in the ratio $2:\pi:4$. Hence the thrust is least in C.V. and greatest in U.A.R.

In practice it is desirable that the maximum pressure angle $\theta$ should not exceed about 30°, and in design it is necessary to find a minimum radius for the pitch curve to ensure this.

The U.V. cam will first be considered, see fig. 4. The base circle radius is R, and the distance AB subtended by the angle $\alpha$ for the lift period (180° in this case) is used for the displacement diagram on the right. Ordinates from this diagram are marked off radially to give the pitch curve. It will be seen that the actual pressure angle is not constant; it is a maximum at A, and $\theta > \theta_1 > \theta_2$.

The value of $\tan\theta$

$$= L \div \frac{2\pi R\alpha}{360};$$

or

$$\tan\theta = \frac{360L}{2\pi R\alpha}. \quad . \quad . \quad (1)$$

Hence

$$R = \frac{360L}{2\pi\alpha\tan\theta}. \quad . \quad . \quad (2)$$

From fig. 2 it will be seen that the slope for the S.H.M. and U.A.R. displacement curves rises from zero to a maximum at half lift, a comparison between the maximum values being given by C.V. : S.H.M. : U.A.R. :: $2:\pi:4$. For S.H.M. and U.A.R. profiles the above formulæ may be taken as:

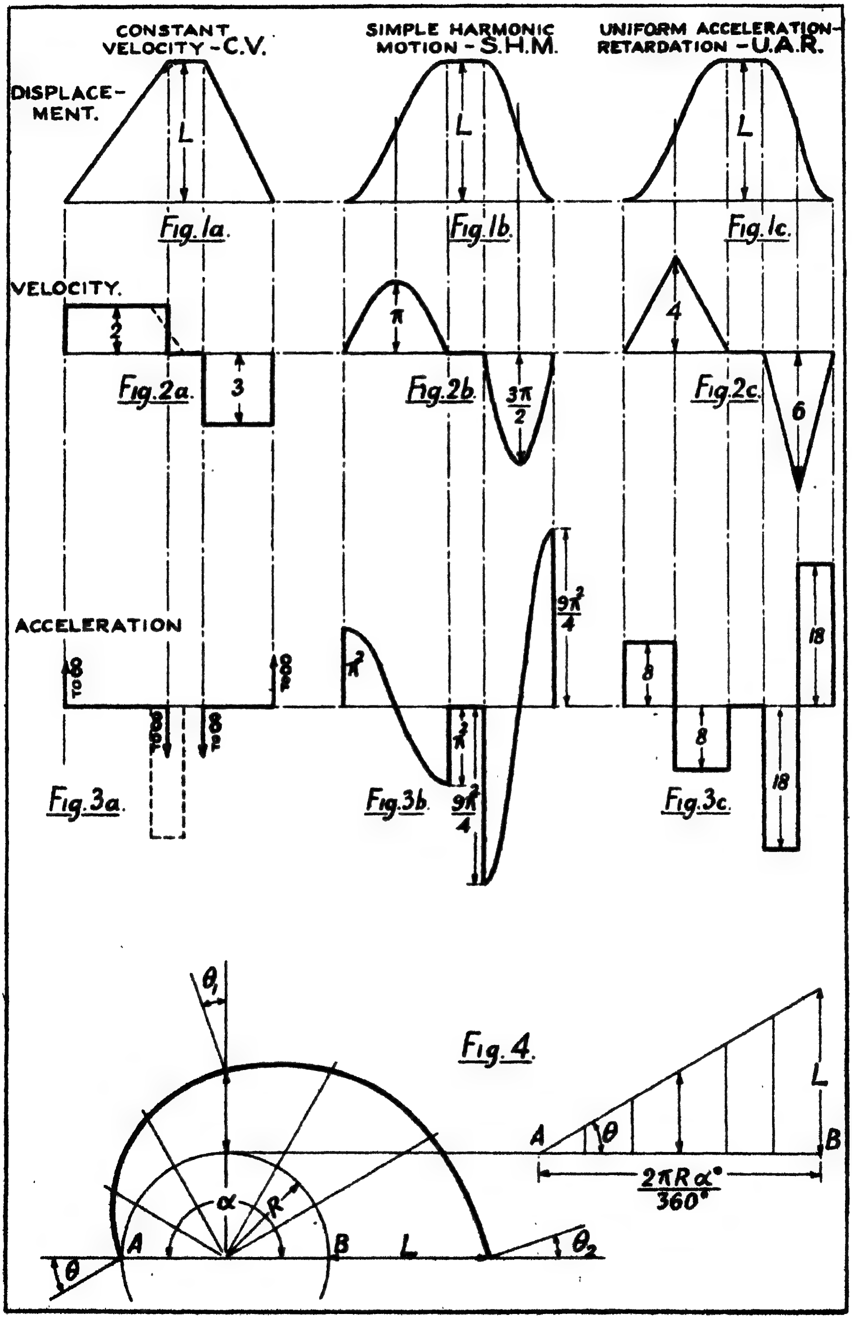
$$\tan\theta = \frac{360\,L\,.\,k}{2\pi\left(R+\frac{L}{2}\right)\alpha}, \quad . \quad (3)$$

and

$$R = \frac{360\,L\,.\,k}{2\pi\alpha\tan\theta} - \frac{L}{2}, \quad . \quad (4)$$

where $k = \frac{\pi}{2}$ for S.H.M. and $\frac{4}{2}$ for U.A.R.

† The graphical construction of velocity and acceleration curves from displacement curves, and conversely, of velocity and displacement curves from acceleration curves, is fully discussed in the author's *Practical Geometry and Engineering Graphics.*


CONSTANT VELOCITY - C.V.
SIMPLE HARMONIC MOTION - S.H.M.
UNIFORM ACCELERATION-RETARDATION - U.A.R.
DISPLACEMENT.
L
L
L
Fig. 1a.
Fig. 1b.
Fig. 1c.
VELOCITY.
2
3
π
3π/2
4
6
Fig. 2a.
Fig. 2b.
Fig. 2c.
ACCELERATION
∞ to
π²
9π²/4
8
8
18
18
Fig. 3a.
Fig. 3b.
Fig. 3c.
Fig. 4.
θ₁
α
R
A
B
L
θ
θ₂
θ
2πRα°/360°

**Cam Design.**—A decision has first to be made on the choice of motion. If this need not be uniform then it can be either S.H.M. or U.A.R. The central alignment of the follower, and the adoption of a roller end are generally to be preferred, but it may be essential to use off-set, flat-ended, or oscillating followers.

The radius of a roller should not exceed the radius of curvature of the pitch curve, where this is convex otherwise these convex parts will give errors in the motion. It is usual to make the roller radius about half the minimum cam radius (i.e. one-third of R).

The total load, including inertia loads, on the working face governs its width. Contact pressures vary between wide limits, but for steel, 500 pounds load per inch width is conservative.

*Example A, Fig. 1.*—Cam speed, 200 r.p.m. Follower motion S.H.M.: rise $2\frac{1}{2}''$ for 140°, dwell 60°, fall $2\frac{1}{2}''$ for 160°. Max. pressure angle 30°.

From (4) on p. 94, R is found to be $1{\cdot}53''$. This is increased to $1\frac{3}{4}''$. For R = $1\frac{3}{4}''$, (3) on p. 94 gives $\theta = 28°$. The roller radius can be $1\frac{3}{4} \div 3$, say $\frac{9}{16}''$, giving a minimum cam radius of $1\frac{3}{16}''$.

The "representative crank" (see p. 92 and opposite) has a radius ($r$) of $1\frac{1}{4}''$. The time of rise takes $(140 \div 360)(60 \div 200)$ sec., i.e. $7 \div 60$ sec. Hence the "crank" revolves at $2\pi(60 \div 7) \div 2$ radians per sec. ($\omega$). Max. vel. is $\omega r = 2{\cdot}8$ f.s.; and max. acc. is $\omega^2 r = 75{\cdot}6$ f.s.s.

In drawing the profile, the cam is supposed to be fixed while the follower revolves about the cam axis in a direction opposite to the rotation of the cam, taking up its required positions. Only the follower end need be shown. The working curve is tangential to the successive roller positions.

*Example B, Fig. 2.*—Cam speed, 1000 r.p.m. Rise $1''$ for 90°, fall $1''$ for 90°, dwell 180°. Maximum pressure angle 30°. Motion (*a*) S.H.M., (*b*) U.A.R.

From (4) on p. 94 R = (*a*) $1{\cdot}23''$, (*b*) $1{\cdot}7''$. They have been assumed (*a*) $1\frac{1}{4}''$, (*b*) $1\frac{3}{4}''$. The roller radii have been taken as (*a*) $\frac{7}{16}''$, (*b*) $\frac{9}{16}''$. Fig. 2 shows S.H.M. on the left, U.A.R. on the right, for half the cam. (The constructions for the points are those of figs. 1 and 3.)

The calculation of max. vels. and accs. is left as an exercise. Note: (p. 95, figs. 2*b* and 2*c*) Vel. U.A.R. = Vel. S.H.M. $\times (4 \div \pi)$; also (figs. 3*b* and 3*c*) Acc. U.A.R. = Acc. S.H.M. $\times (8 \div \pi^2)$.

*Answer.*—Vel.: S.H.M., 8·73, U.A.R., 11·1 f.s.; Acc.: S.H.M., 1830, U.A.R., 1480 f.s.s.

*Example C, Fig. 3.*—Mushroom follower, to rise $3''$ for 90° with U.A.R., to fall $3''$ for 90° with U.A.R., dwell 180°. Least radius of cam $5''$.

One-half the profile is drawn opposite. A straight line representing the flat end of the follower is carried around the cam axis to pass through the plotted points, and the working curve is drawn to touch these lines tangentially. The point of contact moves off the radial line appreciably. E.g. when the cam turns to bring radial line 3 vertical, contact will be at $p3$. This causes a bending moment on the follower—which is increased if it is offset as in fig. 3, p. 93.

## EXERCISES

(1) Complete the profile for Ex. A and completely design the cam for a cam shaft $1\frac{1}{2}''$ diam. Width of cam face, $1\frac{1}{2}''$.

(2) Repeat Ex. A for a U.A.R. cam.

*Answers.*—R, $2{\cdot}3''$; vel. 3·57 f.s.; acc. 61 f.s.s. (See Ex. B.)

(3) Draw the profiles in Ex. B three times full size. Compare them; also show the small deviation between each and a profile which has circular arcs and tangent lines.

(4) Complete Ex. C and find a suitable diameter for the flat end, and the maximum displacement of the line of contact. Assume the centre of the cam face offset $2''$ from the axis of the follower.

(5) Refer to p. 93, fig. 4. The cam rocks through angle $\alpha$ and gives the follower a movement L. The profile is part of the involute (see p. 110) of the base circle, rad. $b$, to which the follower axis is tangential. Extreme positions for the involute are shown dotted. Design such a cam if L = $3''$ for $\alpha = 60°$. Investigate the motion of the follower.

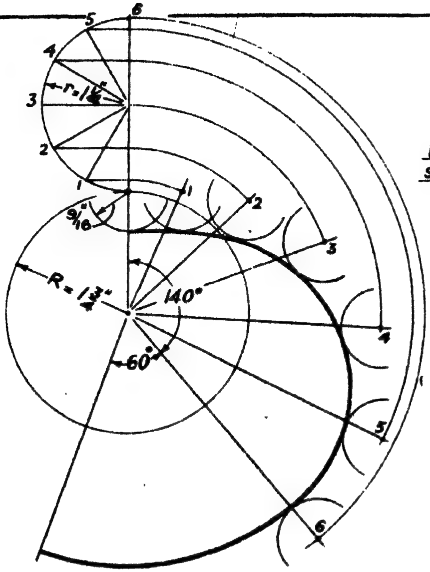



Fig. 1.
scale 3/8.

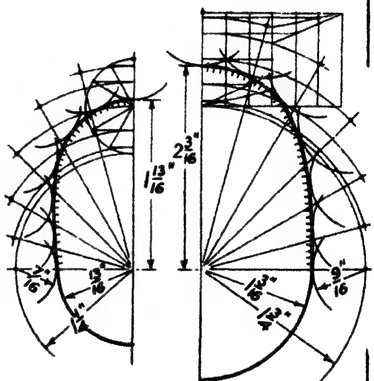



Fig. 2.
scale 1/2.

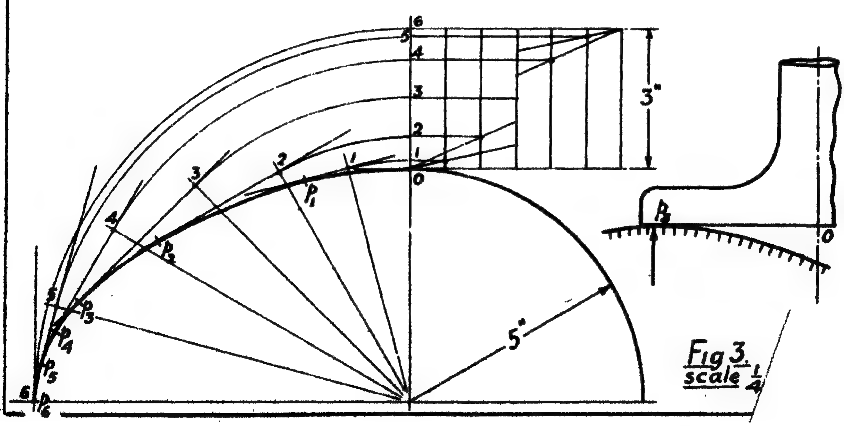



Fig 3.
scale 1/4

*For permissible bearing pressures refer to p. 222 in Appendix*

**Pedestal Bearings for Rotating Shafts. Solid Type.**—The simplest kind of shaft bearing is formed by drilling a hole in a supporting piece to receive the shaft. If the bearing proper is to be distinct from the support, the pedestal design shown in fig. 1 may be adopted: the proportions of the various parts are given in terms of the shaft diameter D. The hole will eventually wear oval, and to enable adjustment to be made, the bearing is usually lined with a thin G.M. bush: the provision of a new bush restores the bearing to its original condition. The bush is usually simply pressed into the hole in the pedestal. For small bearings friction alone is relied upon to keep the bush in position, but set screws or dowel pins must be fitted to prevent large bushes from moving.

**Split Type.**—A more satisfactory method of taking up wear is by dividing the bearing by a plane containing the axis of the shaft and normal to the direction of the load. In fig. 2 the upper half or cap is secured to the base or pedestal by means of studs. Alignment between the parts is maintained by a narrow offset A on the line of division: to ensure that no *longitudinal* relative movement occurs, the offset portion in the cap need not extend the whole length and may then fit into a *recess* arranged in the lower half. Wear is taken up by removing some of the metal at B:* the bottom of the offset at A is usually clear of the pedestal.

Important bearings are invariably lined either with whitemetal (cast in the halves of the bearing and then machined) or with split gunmetal bushes. Both methods are discussed later.

**Footstep or Thrust Bearing.**—The simple type shown in fig. 3 is suitable only for a slow-running lightly loaded shaft. If the shaft is not of steel, its end must be fitted with a steel face. This end is rounded, and is supported on a cup-shaped steel disc fitting in the footstep and prevented from rotating by a steel pin A. The shaft is guided in a gunmetal bush, pressed into the pedestal and prevented from turning by the pin B.

This type of bearing has two objectionable features: (1) it is difficult to lubricate, the oil being thrown outwards from the centre by centrifugal force; (2) the wear on the disc is not uniform over its surface, the rubbing velocity varying from a minimum at the outer edge to zero at the centre.

**Ball bearings** are widely used to take axial loads applied to small shafts; and for large shafts the **Michell type of thrust bearing** is now almost exclusively adopted. Both bearings are fully described later.

## EXERCISES

*The following are all to be drawn full size and dimensioned*

(1) Draw an elevation, plan, and sectional end view of a bushed pedestal bearing suitable for a shaft 2″ diameter.

(2) Draw the following views of the C.I. bracket and bearing shown in fig. 2: elevation (i.e. a view along the shaft axis) showing a portion of the bracket in section through a stud, as in figure; an end view; a plan. In the last two, show the *complete* pedestal. Assume that the bearing overhangs the base symmetrically, and use your own judgment in proportioning undimensioned parts.

(3) Draw an elevation, half in section, and a plan of a simple footstep bearing for a shaft 2½″ dia.

* Thin metal strips called shims are frequently provided initially at B, fig. 2. Wear may then be taken up by removing one or more of the strips.

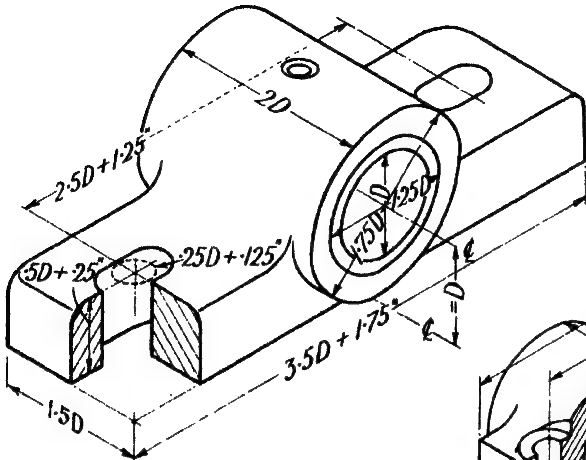



Fig. 1
C. I. PEDESTAL BEARING, WITH G. M. BUSH.

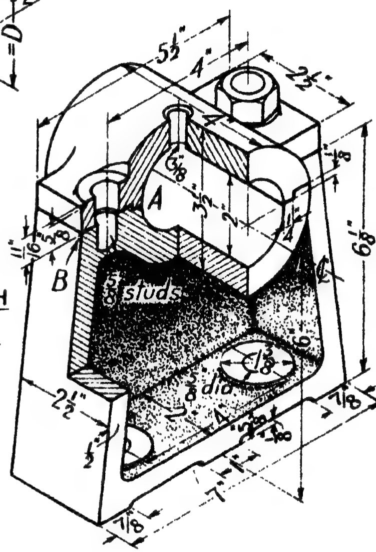
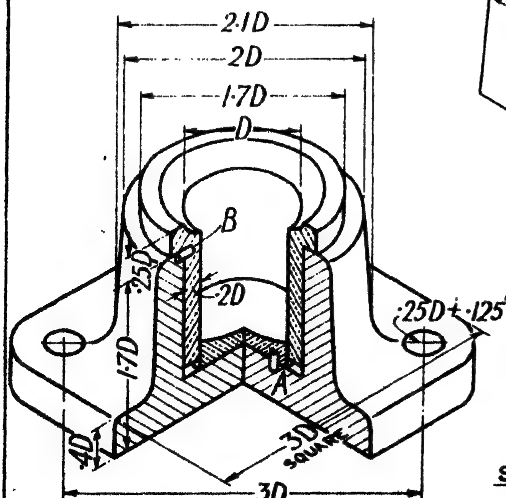



Fig. 2
C. I. BRACKET WITH SPLIT BEARING



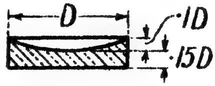
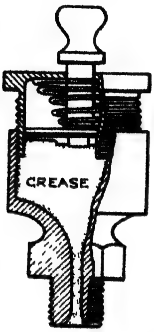




Fig. 3.
SIMPLE FOOTSTEP BEARING.

**A Simple Pedestal Bearing or Plummer Block,** suitable for a 2" dia. lightly loaded shaft, is shown opposite. The shaft is carried in G.M. half bearings or **steps,** fig. 1, which are supported or housed in a C.I. **pedestal,** fig. 4. A boss on the upper step enters a hole in the **cap,** figs. 2 and 3, and prevents relative movement: the boss also serves for the attachment of a Stauffer grease lubricator of the kind shown in the annexed figure. With the exception of the central recessed portion, the steps are machined all over. The halves butt when in place; the cap, however, when pressing on the upper step, does not bottom in the pedestal.




In order that a design may be competitive commercially all unnecessary metal must be eliminated and excessive machine work avoided. These factors have been studied closely in the production of the pedestal and cap shown; both parts are economically proportioned and require machining only at the step bearing surfaces, and on the parts of the cap and pedestal in contact. The base is recessed on the underside to produce a bearing strip which is finally ground true.

**Proportions.**—The diameter and length of the bearing govern all other dimensions, and for a given diameter the length depends entirely on the load carried. For grease-lubricated bearings a pressure $p$ of 30–50 lb./in.$^2$ of *projected* area is permissible;* i.e. $p$ = load in pounds $\div$ (length in inches $\times$ diameter in inches). The length may vary from 1½ to 2 $\times$ diameter. The following table gives important dimensions found to be suitable in practice.

(Note.—Suitable brackets for supporting this pedestal are shown on pages 168 and 169).

Dimensions in Inches.

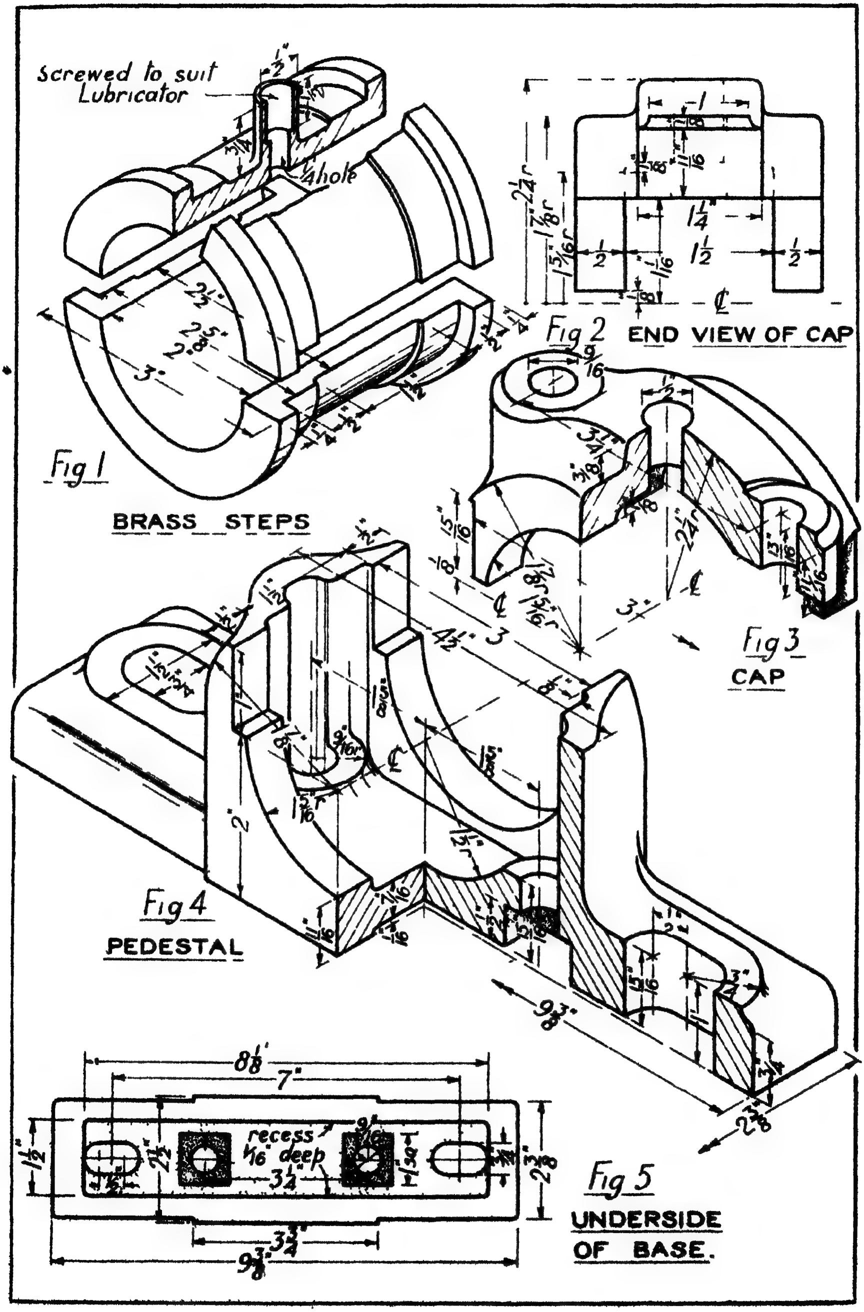
| Shaft Dia., D. | Steps Thickness, Max. | Pedestal Height. | Axis of Shaft, Height = D. | Cap. | | | | Base. | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | Bolts. | | | Thickness, Min. | Width. | Thickness, Max. |
| | | | | No. | Dia. | Centres. | | | |
| 1½ | ¼ | 2¼ | 1½ | 2 | ⅜ | 2⅝ | ⅝ | 1¾ | 13/16 |
| 2 | 5/16 | 3 | 2 | 2 | ½ | 3¼ | 11/16 | 2⅝ | 1 |
| 2½ | 5/16 | 3¾ | 2½ | 2 | ⅝ | 4 | 1 1/16 | 3 | 1¼ |
| 3 | ⅜ | 4⅜ | 3 | 2 | ¾ | 4⅞ | 1 3/16 | 3¾ | 1⅜ |
| 3½ | ⅜ | 5 | 3½ | 2 | ⅞ | 5⅝ | 1⅜ | 4⅝ | 1¾ |
| 4 | ½ | 5½ | 4 | 2 | 1 | 6⅜ | 1½ | 5¾ | 2 |
| 5 | ⅝ | 6¾ | 5 | 2 | 1⅛ | 7⅞ | 2 | 7 | 2⅜ |
| 6 | ¾ | 8 | 6 | 4 | 1⅛ | 9⅜ | 2¼ | 8¼ | 2¾ |

## EXERCISES

(1) Draw, full size, the following views of the pedestal bearing shown, with the various component parts assembled; elevation, half in section through the bolts; end view, half in section on the shaft axis; plan, one half showing the cap and steps removed. Dimension the views and insert finish marks. Omit a lubricator.

(2) Prepare, full size, complete working drawings of a bearing suitable for a 3" dia. shaft, similar to the bearing shown opposite, with steps 4½" long.

* In the Sellers' cast-iron bearing for factory shafting the bearing pressure $p$ was limited to 15 lb./in.$^2$.

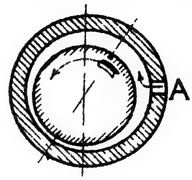

Screwed to suit Lubricator
1/4 hole
Fig 1
BRASS STEPS
Fig 2
END VIEW OF CAP
Fig 3
CAP
Fig 4
PEDESTAL
recess 1/16" deep
Fig 5
UNDERSIDE OF BASE.

**Ring-oiled Shaft Pedestal.**—In this very satisfactory type of bearing, lubrication is effected by rings which hang upon the shaft through gaps arranged in the upper step, figs. 1 and 2, and which raise oil as they revolve from an oil-well formed in the lower part of the pedestal, fig. 4. This system of lubrication is widely used and should be adopted in all but the simplest of shaft bearings. The oil-level indicator and drain cock usually provided for bearings of this type have been omitted to avoid complication. Oil is prevented from creeping along the shaft by the provision of oil throwers: these consist of narrow rings, usually of triangular section, secured to the shaft within the pedestal case.

Details of a ring-oiled **Swivel Bearing and Pedestal** are given on pp. 188 and 189.

The **proportions** are largely empirical and may be arrived at without much difficulty by suitably modifying the dimensions given on p. 100 Higher pressures and speeds are permissible in these bearings than in those described on previous pages. The length of the steps varies from 2 to 3 × diameter of shaft.

**Oil-film Lubrication.**—It has been demonstrated that, providing there is a finite difference between the radii of a journal and its bearing, a separating oil film can be permanently maintained on the surfaces.* In the figure the clearance has been exaggerated. Oil entering at A is carried around by the journal and is compressed as it passes into the wedge-shaped space. If the speed is great enough, the pressure generated will be sufficient to lift the shaft, thus permitting the formation of a continuous




oil film. The pressure distribution in the film balances the load, and the only frictional resistance is that occasioned by the molecular shearing action in the lubricant.

It will be evident, therefore, that if an adequate supply of lubricant can be introduced at the point of least pressure, the wear on the bearing should be negligible. This is the case with well-designed bearings working under *forced lubrication*, wear occurring only on starting up and slowing down. Modern practice with these bearings is to reduce the length, making it equal to the diameter of the shaft; to dispense with oil grooves; and to operate with just sufficient oil to keep the temperature at about 140° F.†

The seizing of bearings depends not only upon the intensity of pressure but also on the rate at which the heat generated by friction can be dissipated. In Tower's experiments, under the most favourable conditions, seizing occurred at mean pressures of about 600 lb./in.$^2$; this figure has, however, been greatly exceeded, without seizing, in more modern researches.‡

## EXERCISES

(1) Draw, full size, the following views of the bearing shown, with the various component parts assembled: sectional elevation on the shaft axis; end elevation, one half in section at a ring; plan. Insert overall dimensions.

(2) Prepare fully dimensioned working drawings of a ring-oiled bearing suitable for a shaft 3″ dia. Use steps 4½″ long.

* Beauchamp Tower's researches, carried out for the Frictional Research Committee and reported in *Proc. I. Mech. E.*, 1885. Subsequently Osborne Reynolds took up the question from the theoretical side and deduced a hydrodynamical basis for the phenomena.
Refer also to Charnock's paper "Bearings for Line Shafting", *Proc. I. Mech. E.*, 1929.

† "Journal Bearing Practice", *Proc. I. Mech. E.*, 1929.

‡ "Recent researches on Lubrication", *Proc. I. Mech. E.*, 1922. In the experiments reported, bearing pressures of 7840 lb./in.$^2$ were maintained.

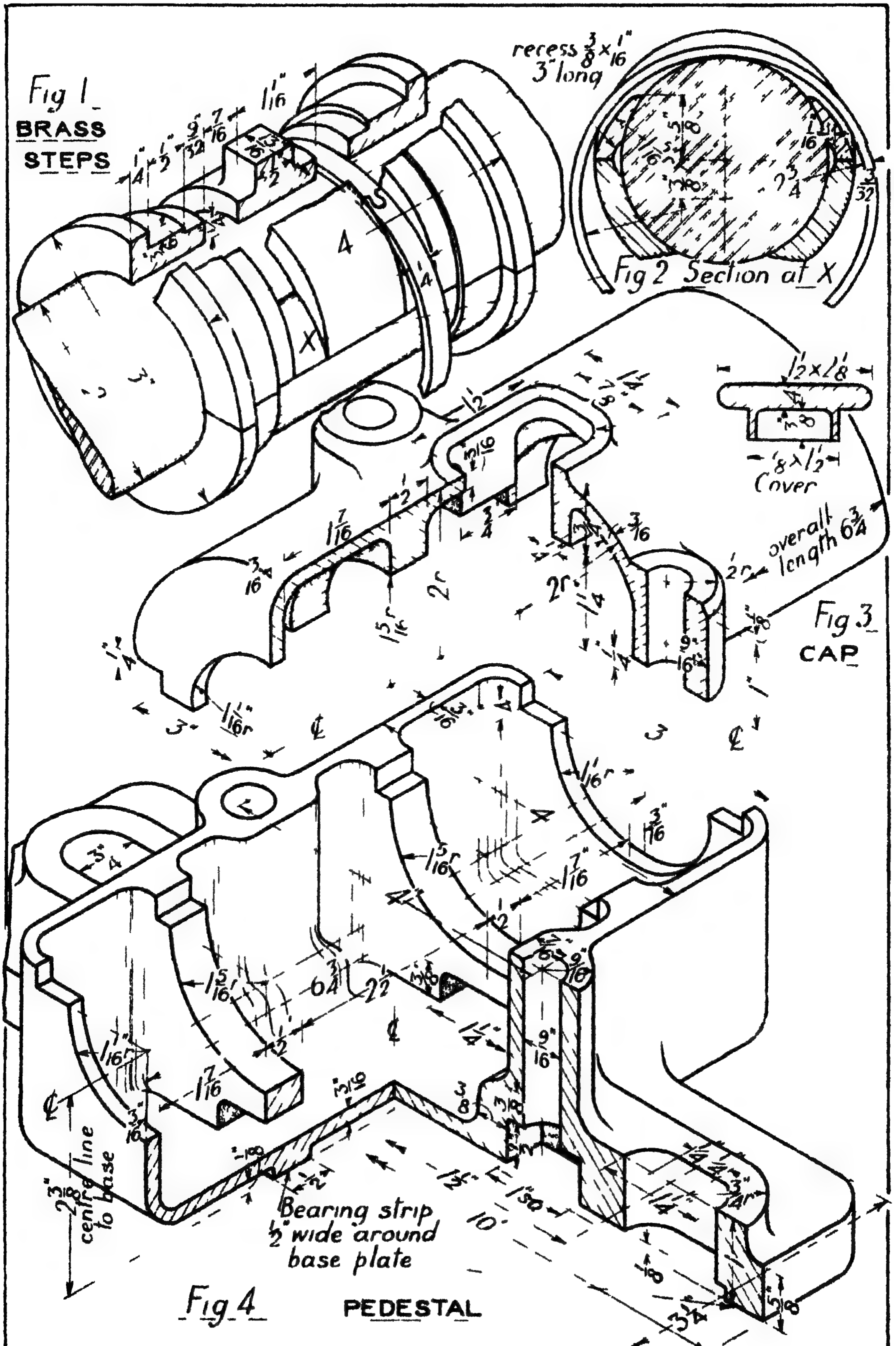


Fig 1
BRASS STEPS
Fig 2 Section at X
recess 3/8 x 1/16 3" long
Cover
overall length 6 3/4
Fig 3
CAP
Bearing strip 1/2" wide around base plate
centre line to base
Fig 4
PEDESTAL

**Whitemetalled Bearings.**—Certain alloys of tin, copper, and antimony form better bearing surfaces than C.I. or G.M., giving a reduction in friction and wear. They are named after the inventor of the first successful alloy, Babbitt. The proportions of Babbitt's bearings are given as: tin 87 per cent, copper 7·8 per cent, antimony 5·2 per cent, but authorities differ as to the exact proportions. Cheaper alloys of lead and antimony are now commonly used, and the name whitemetal (W.M.) covers all variations.

W.M. is too weak to be used other than as a lining in steps of C.I. or G.M. The molten metal is usually poured into the step around a mandril slightly smaller than the journal. After cooling, the W.M. is scraped or machined to size. If a W.M. bearing becomes overheated the fusible alloy melts and runs out.

The metal has a tendency to flow or "creep" under pressure, and it is necessary to anchor the lining to the step. This is usually effected by the provision of dovetailed recesses in the step, as shown opposite. A bearing pressure of 300 lb./in.$^2$ is commonly adopted.

**Turbine Shaft Bearings.**—The dropping of a turbine shaft consequent upon the overheating and "running" of a W.M. bearing would probably have disastrous results on the blading. For these bearings wide G.M. bearing strips, arranged about 0·015″ below the W.M. surface, are provided to support the shaft in such an emergency. The strips are shown in fig. 2.

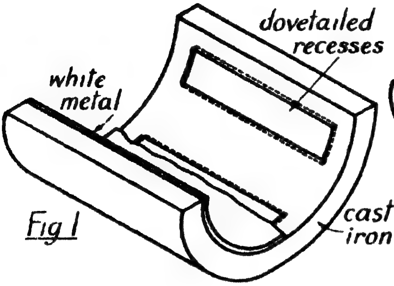
**Crank Pin Bearing.**—Figs. 3, 4, and 5 show the construction of a large whitemetalled bearing suitable for the crank pin of a land-type steam-engine. The steps are of C.S. and the dovetailed grooves are cast in. The halves are separated by G.M. liners which permit adjustment due to wear. Oil recesses ⅛″ deep are cut in the W.M. at the top and bottom and are connected to an oil hole by shallow grooves. This bearing is designed for the large end of the connecting rod shown on pages 194 and 195, to which reference should be made. The actual design has been slightly simplified in fig. 3 to form a drawing exercise: e.g. the G.M. liners are not quite as shown (refer to page 194); further, the C.S. steps require cutting away at top and bottom to clear the connecting bolts.

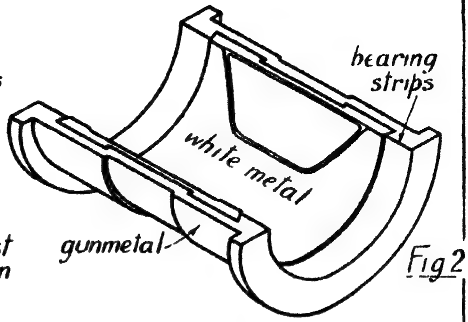
**Railway Carriage Axle Bearing.**—An example of a successful type of W.M. bearing is given on page 184, showing the axle bearing of the L. M. & S. Railway stock. A half-bearing only is required, and the actual bearing surface is reduced by easing away the metal at the sides. The oil is conveyed to the bearing by a lubricating pad, immersed in an oil bath and pressing on the underside of the axle. The maintenance of a continuous oil film between the surfaces practically eliminates wear. Bearings of this type show a mirror-like surface after 150,000 miles of running, and in actual comparative tests on rolling stock they were found to be equally as effective as roller bearings.* The bearing pressure allowed is 300 lb./in.$^2$.

## EXERCISE

Draw, preferably half size, but if a limited space is available one-third size, the following views of the bearing shown in fig. 3: elevation, taken in direction of the arrow A, showing the left-hand half in section on the centre line; plan, showing the left-hand half in section *on the centre line*; end view, on the joint. Dimension the views.

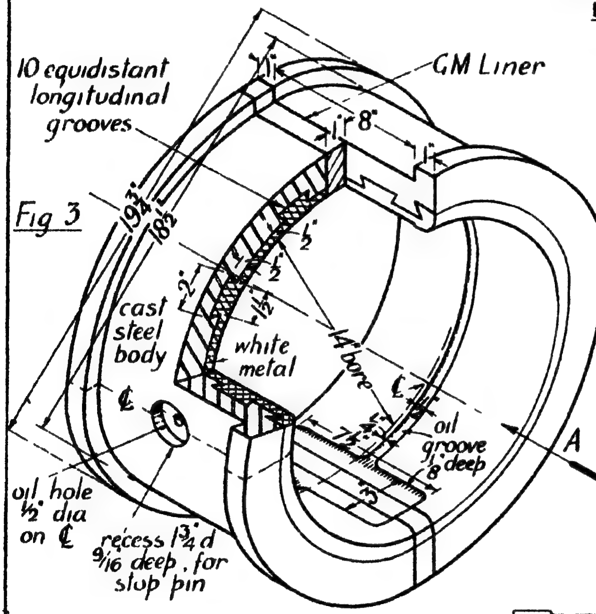
* "Anti-friction Bearings", *Proc. I. Mech. E.*, Feb., 1925. Roller bearings, of course, would have the advantage at starting.

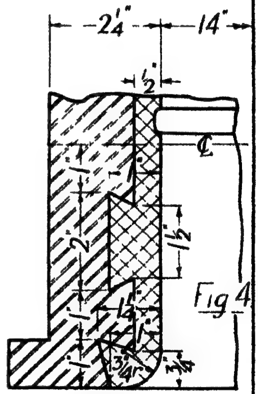



WHITE METALLED HALF-BUSH

TURBINE SHAFT BEARING

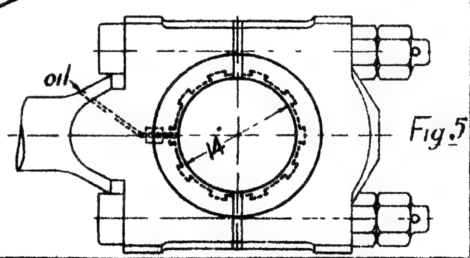



WHITE METALLED BEARING

FOR

CONNECTING ROD END



**Ball Bearings** have the following advantages over plain bearings:—the coefficient of friction is lower ($\mu = 0{\cdot}0012$) and *is practically the same at starting as in motion*; heavier loads and higher speeds are permissible; less space is required; less lubricant is used; and wear is practically nil. On the other hand, the first cost is greater, the bearings cannot be used in halves, any defects may produce serious consequences, and the bearings are frequently noisy after long use.

Two of the most common of the many forms of ball bearings are shown opposite, viz. a **journal bearing**, fig. 1, and a **thrust bearing**, fig. 2. The races for each are grooved, the radius of the groove being from 5 per cent to 10 per cent greater than the radius of the ball. Both balls and races are of high-carbon chrome steel, hardened and polished. The balls are parted to prevent contact friction by means of cages made of anti-friction alloy. The journal bearing cage, fig. 1, is in halves riveted together, with radial holes to take the balls. The balls are introduced as shown in fig. 3; the race rings are then spaced concentrically and the halves of the cage inserted from each side. In the thrust bearing cage, fig. 2, the holes are drilled to a shape which permits the entry of the balls from one side; the projecting central ridge is then indented and closed over sufficiently to retain the balls.

**Proportions.**—The manufacture of ball bearings is a speciality, and the machine designer usually has only to select a bearing suitable for the particular shaft diameter, speed, and load—the manufacturers giving essential dimensions as in fig. 1—and to design a housing for it.

Dimensions of standardized Ball Bearings and Parallel-Roller Bearings are given in B.S. 292.

The following formula, due to Goodman, gives the safe load P for a ball bearing of given dimensions:

$$P = \frac{k \cdot m \cdot d^2}{nD + cd} \text{ lb.},$$

where D = ball centre diameter in inches, $d$ = ball diameter in inches, $m$ = number of balls in bearing, $n$ = shaft speed in revs./min.

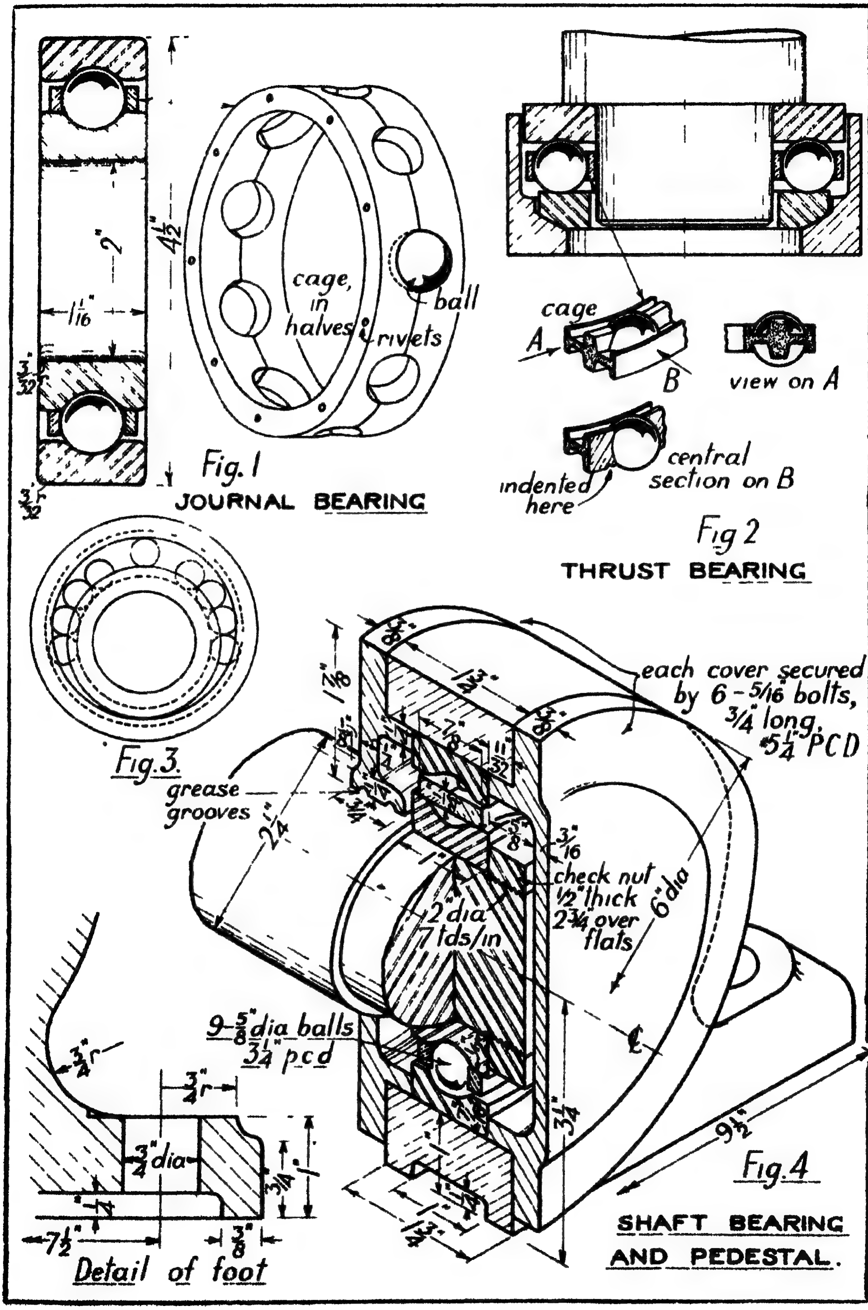
Values of the constants $k$ and $c$ are as under:

| Type of Bearing | $k$ | $c$ |
|---|---|---|
| Radial, flat races - | 1,000,000 | 2000 |
| Radial, hollow races | 2,500,000 | 2000 |
| Thrust, flat races - | 500,000 | 200 |
| Thrust, hollow races | 1,250,000 | 200 |

In many cases manufacturers quote higher values for the safe load than that given by the above formula.

## EXERCISE

A location bearing for the end of a shaft is shown in fig. 4, the pedestal being symmetrical about the central section. The design incorporates the bearing shown in fig. 1: the inner race is pressed on the shaft and secured by a check nut, and the outer race is held by the spigots of the end flanges. Draw, full size, the following views and dimension them: elevation (i.e. a view along the axis), with cover removed; end view; sectional end view on shaft axis. Show a locking arrangement for the nut and provide for a Stauffer lubricator on one cover. Calculate the safe load for the shaft, taking a speed of revolution of 1000 revs./min. *Refer also to pages* 196 and 197.


cage, in halves & rivets
ball
2"
4 1/2"
1 1/16"
3/32"
Fig. 1
JOURNAL BEARING
cage
A
B
view on A
indented here
central section on B
Fig 2
THRUST BEARING
Fig. 3.
grease grooves
2 1/4"
1 7/8"
4 3/4"
each cover secured by 6 - 5/16 bolts, 3/4 long, 5 1/4" PCD
check nut 1/2" thick 2 3/4" over flats
6" dia
2 dia 7 tds/in
9-5/8" dia balls 3 1/4" pcd
3 1/4"
9 1/2"
Fig. 4
SHAFT BEARING AND PEDESTAL.
3/4" r
3/4" dia
7 1/2"
3/8"
Detail of foot

**Roller Bearings.**—A great variety of types of roller bearings have been produced, and advantages have been claimed for many of them which have not always been borne out in practice. The earlier designs, in which long mild-steel rollers, both cylindrical and tapered, were used, proved unsatisfactory under load at high speeds—principally because it was found impossible to keep the axes of the rollers parallel to the shaft axis. Modern practice is to use short rollers of hardened high-carbon chrome steel; in the type shown opposite, the length is made equal to the diameter.

**Roller Journal Bearing. Figs. 1, 2, and 3.**—The rollers are solid cylinders with the ends perpendicular to the axes. They roll around a grooved channel in the inner race and are spaced and kept parallel to the shaft by means of a cage of substantial proportions. The construction of the cage should be clear from figs. 2 and 3. The effect of the rubbing action between the rollers and the cage is not appreciable. Both races must be secured longitudinally, and the revolving race (usually the inner) is an interference fit on the shaft. The bearing shown will carry from 50 per cent to 70 per cent more load than the ball bearing shown on the previous page, at the same speeds.

Details of standardized Parallel-Roller Bearings are given in B.S. 292.

**Self-aligning Bearing to carry journal load and end thrust. Fig. 4.**—The complete pedestal is symmetrical about the central section. The roller-bearing unit shown in fig. 1 is incorporated in this design but the outer race is made spherical, externally, and is free to turn in a shell, bored spherically and made a sliding fit in the bore of the pedestal. The inner race grips the shaft and is clamped endways by the flanged sleeves—upon which the couplings press. In conjunction with the roller bearing are mounted two single-thrust bearings, the stationary races of which are provided with spherical seatings. The three bearings swivel about a common centre. The design is suitable for use as a marine thrust block, and the thrust bearings are capable of taking the following loads:—

| | | | | | |
|---|---|---|---|---|---|
| Speed, revs./min. | 100 | 300 | 600 | 1000 | 2000 |
| Safe load, lb. | 3650 | 2110 | 1490 | 1150 | 810 |

**Other Types of Roller Bearings.**—The Cooper roller bearing is of the split type and is illustrated on pp. 190 and 191; the design overcomes the disadvantage of threading the ordinary kind on the shaft, but at the expense of split races.

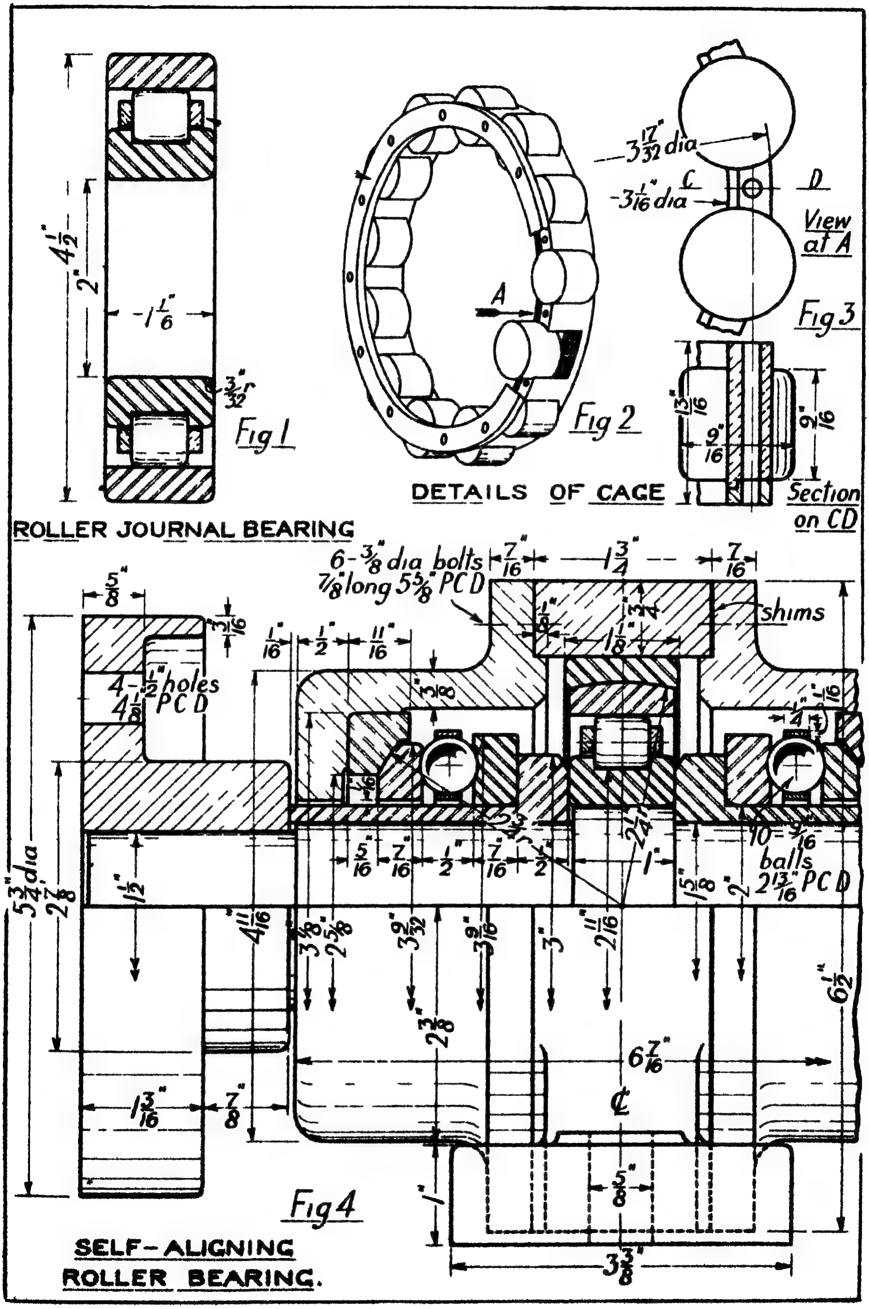
For railway-carriage axles and other heavy-duty bearings in which an end thrust as well as a journal load has to be taken, a design employing short barrel-shaped rollers, bearing on a spherical race track, has proved satisfactory. For details of this bearing, and for a discussion on anti-friction bearings in general, the student is referred to the *Proc. I. Mech. E.*, No. 2, 1925.

**Needle-roller Bearings.**—These have latterly been used with some success. As the name implies, the rollers are needles, of length often as much as 9 times the diameter, not spaced by cages, and not requiring special races if the rotating parts are case hardened and ground. The bearings are useful where the diameter of the housing is restricted or where a saving in weight is essential. They are, however, more expensive, and offer greater frictional resistance, than roller bearings of the ordinary type.

## EXERCISE

The shaft shown in fig. 4 carries a similar coupling on the right, and the complete design is symmetrical about the central section. Draw, full size, the following views of the complete unit: elevation, with upper half in section; plan; end view with the near coupling removed. Dimension the drawing. Use your own judgment where dimensions have been omitted.

*Note.*—The design of a suitable base for the pedestal is left as an exercise for the student. The base is integral with the central barrel and is widest at the bolt holes.



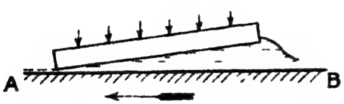

ROLLER JOURNAL BEARING

DETAILS OF CAGE

SELF-ALIGNING ROLLER BEARING.

**Michell Thrust Bearing.**—For shafts transmitting an axial thrust, e.g. propeller shafts of ships, the Michell type of thrust bearing is now widely used, particularly in naval vessels. The bearing itself is a speciality and the mathematical theory underlying its operation is advanced, but in view of its importance a simple design will be dealt with here.*

**Lubrication of Flat Surfaces.**—An oil film cannot be maintained between two perfectly flat and parallel surfaces which have relative sliding motion under a steady load. The oil is squeezed out and the surfaces make physical contact. If, however, the surfaces are slightly inclined to each other, as in the figure below, an oil film can be




maintained. The moving surface AB carries oil into the wedge-shaped space, and a pressure is generated within the film sufficient to sustain a considerable load. In the Michell bearing one thrust surface consists of a number of blocks which are free to tilt when under load, the back of each block being cut away, as shown clearly in fig. 4.

**Michell Bearing Unit.**—The bearing unit only is shown opposite; the housing in which it is held is shown on pp. 198 and 199. A collar on the shaft (not shown) bears against four blocks, one of which is removed in fig. 2, and the axial thrust is transmitted through these to the shoe, and thence to the fixed housing. Two shoes are cut from a complete ring, so that each is slightly less than a semicircle. Each block is an easy fit in the grooved shoe and is prevented from rotating with the collar by a pin which projects into a hole in the block. The back of each block is machined away from a pivoting edge, CD, displaced a definite distance from the geometrical centre. When the shaft is revolving, each block tilts about this edge until equilibrium is reached, the fluid pressure in the oil film between the collar and the blocks then balancing the load.

The bearing shown is suitable for one direction of rotation only: for a reverse direction the blocks would require to be pivoted to the opposite "hand". If the shaft is to transmit thrust in both directions, a similar unit is necessary on the other side of the collar on the shaft.

The bearing is immersed in an oil-bath, and oil carried around by the collar is removed automatically by scrapers and deflected on to the blocks.

## EXERCISES

(1) Prepare a working drawing of one block, as in fig. 4, twice full size. Give (*a*) a view on the working face, (*b*) a view on the back, (*c*) a section on AB, (*d*) an end view from the right on a central radial section.

(2) Draw, full size, the following views of the complete bearing (fig. 2); elevation, on the blocks; plan; end elevation. Dimension the shoe only.

(3) If the limiting bearing pressure is 200 lb./in.$^2$, calculate the maximum thrust load for the bearing.

*Answer.*—2430 *lb.*

* Without the Michell thrust block it would have been impossible to produce large steamships of great horse-power. For example, in one vessel, the thrust of each shaft, amounting to over 100 tons, is taken on one collar: the area of the thrust surface is 1176 in.$^2$.

The mathematical theory was first given by Anthony G. M. Michell in the *Zeitschrift für Mathematik und Physik* in 1905. The theory is fully discussed in *Engineering*, 20th Feb., 1920.

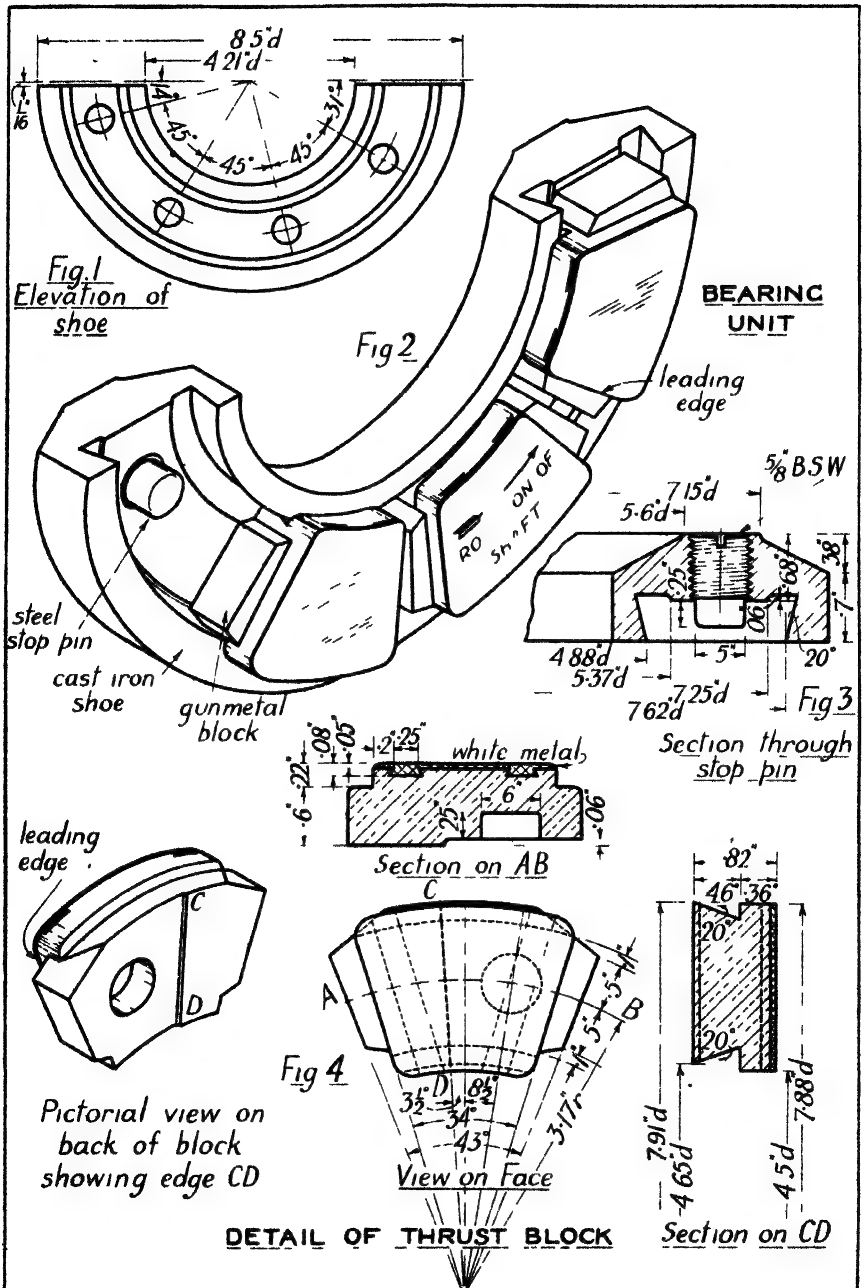


Fig. 1
Elevation of shoe
Fig 2
BEARING UNIT
leading edge
steel stop pin
cast iron shoe
gunmetal block
ROTATION OF SHAFT
5/8 BSW
Fig 3
Section through stop pin
white metal
Section on AB
leading edge
Pictorial view on back of block showing edge CD
Fig 4
View on Face
Section on CD
DETAIL OF THRUST BLOCK

**Involutes.**—If a straight line roll without sliding upon a circle, fig. 1, the locus of any point on the straight line is an involute of the circle.

**To Draw the Involute.**—Apply a straight line drawn on tracing-paper over the base circle (fig. 2); let Q be the generating point and A the point of contact. Using a pricker, mark the position of Q. Then transfer the pricker to A and allow the line to turn about A until it overlaps the circle slightly (dotted line), cutting it at B. Transfer the pricker to B and revolve the line until it is now tangential to the circle (chain line). The point Q has now moved to $Q_1$. Mark $Q_1$: in the same way plot other points and join them with a fair curve.

**Toothed Gearing.**—Two plain wheels A and B, fig. 3, are in contact and revolve about parallel axes, one transmitting motion to the other by friction at the rubbing surfaces. To prevent slipping at P when power is to be transmitted, grooves may be cut in the surfaces and projecting strips added between the grooves, forming the gear teeth shown in fig. 4. The imaginary circles in fig. 4 corresponding to A and B are called the **pitch circles** of the gear wheels, and P is called the **pitch point**; the **pitch** of the teeth is defined on the following page. The height of a tooth above the **pitch surface** is called the **addendum**; the depth below, the **dedendum**, fig. 5. The difference between the addendum of a gear and the dedendum of its "mating" gear is termed the **clearance**.

The profiles of the teeth will be correct when the motion transmitted is the same as that given by the plain wheels in rolling contact; i.e. when the angular velocity ratio is constant. For this it is necessary, and sufficient, that the common normal to the teeth profiles should always pass through the pitch point, and this condition is satisfied if the profiles are either of **involute** or **cycloidal** * form. For reasons given later, the involute form is almost exclusively used at the present time, and only this type will be discussed herein.

In involute teeth the path of the point of contact, called the **path of contact**, is a straight line passing through the pitch point and tangential to the base circles of the involutes forming the teeth profiles. In fig. 6 the teeth have contact at Q, and the locus of Q coincides with the normal $NN_1$ and passes through P, the pitch point. The involute base circles are tangential to $NN_1$ and concentric with the pitch circles. The angle which $NN_1$ makes with $TT_1$, the common tangent at P, is called the **obliquity** or **pressure angle** $\psi$ (*psi*). The points of tangency, N and $N_1$, are called **interference points**: true involute contact cannot extend beyond them.

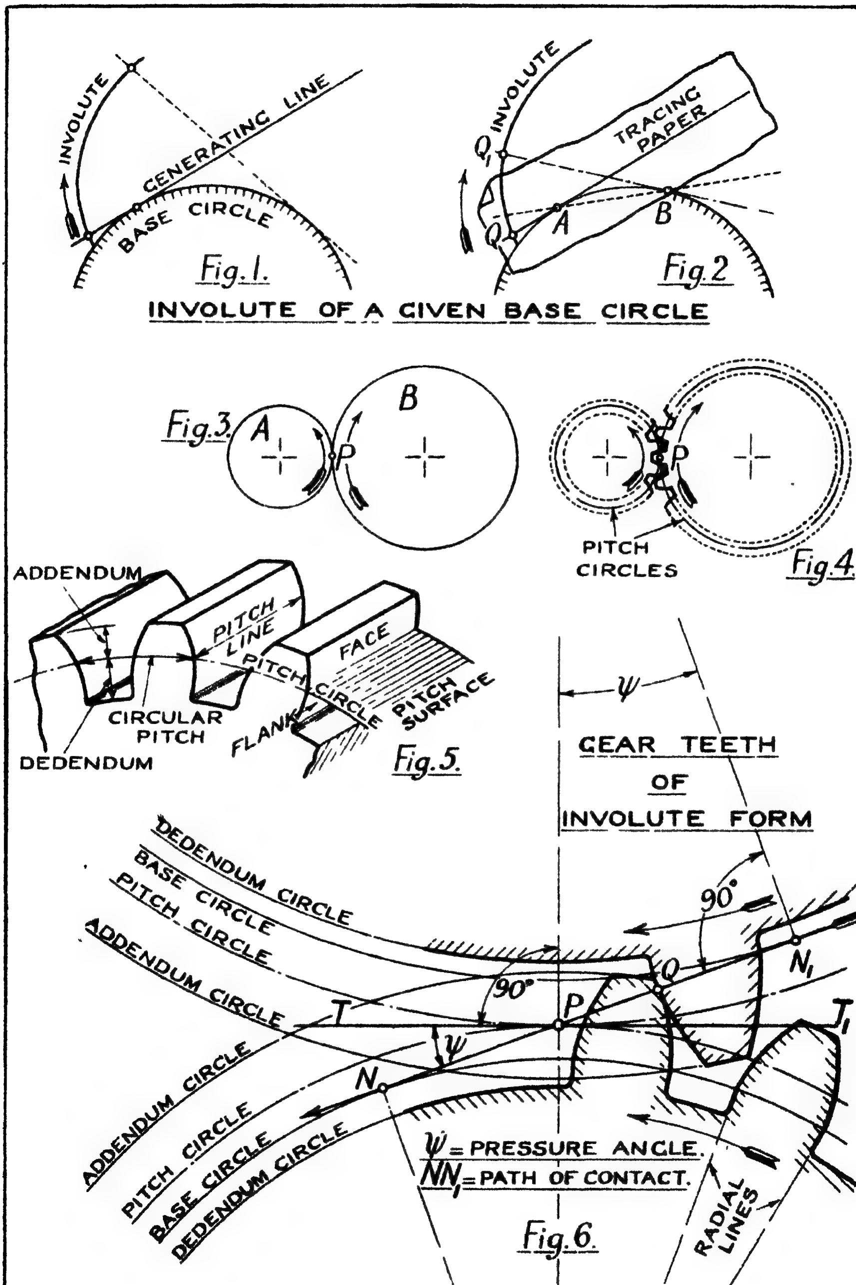
**To draw involute teeth.**—Let the two pitch circles be given (fig. 6). Through P, the pitch point, draw a line $NN_1$ making an angle $\psi$ with the common tangent $TT_1$ ($\psi = 14\frac{1}{2}°$ or $20°$).† Draw circles concentric with the pitch circles and tangential to $NN_1$: *these are the base circles of the involutes.* Using tracing-paper, plot short portions of involutes to the base circles. The addendum and dedendum circles limit the involutes. The flanks of the teeth *within* the base circles are not working parts and may have any form which does not interfere with the faces of the other teeth; usually their profiles are radial lines.

## EXERCISE

The pitch circles of two gears are 10″ and 16″ dia. Taking $\psi = 20°$, determine the shape of one tooth for each gear, full size, making the circular pitch = 2·1″ (width of tooth 1·05″), addendum = 0·66″, dedendum 0·77″. Cut out the teeth in cardboard, using strips long enough to include the centres, and show (1) that the path of contact is a straight line passing through the pitch point, (2) that both rolling and sliding contact occur between the teeth.

* Cycloidal curves are fully discussed in the author's *Practical Geometry and Engineering Graphics.*

† In one of the earliest systems, that of Brown and Sharpe, R.I., U.S.A., $\psi$ was taken as $14\frac{1}{2}°$.


INVOLUTE
GENERATING LINE
BASE CIRCLE
Fig. 1.
INVOLUTE
TRACING PAPER
$Q_1$
$Q$
A
B
Fig. 2
INVOLUTE OF A GIVEN BASE CIRCLE
Fig. 3.
A
B
P
P
PITCH CIRCLES
Fig. 4.
ADDENDUM
PITCH LINE
FACE
PITCH CIRCLE
CIRCULAR PITCH
DEDENDUM
FLANK
PITCH SURFACE
Fig. 5.
$\psi$
GEAR TEETH OF INVOLUTE FORM
DEDENDUM CIRCLE
BASE CIRCLE
PITCH CIRCLE
ADDENDUM CIRCLE
90°
90°
Q
$N_1$
P
T
$T_1$
$\psi$
N
ADDENDUM CIRCLE
PITCH CIRCLE
BASE CIRCLE
DEDENDUM CIRCLE
$\psi$ = PRESSURE ANGLE.
$NN_1$ = PATH OF CONTACT.
RADIAL LINES
Fig. 6.

**Pitch—Circular, Diametral, Module.**—The pitch of teeth may be defined in one of three ways:

(*a*) **Circular Pitch,** $p$, the distance between similar points on adjacent teeth, measured along the pitch circle.

(*b*) **Diametral Pitch,** P, the ratio, number of teeth/pitch diameter in inches, i.e. the number of teeth per inch of diameter.

(*c*) **Module Pitch,** $m$, the ratio, pitch diameter in inches/the number of teeth; i.e. the reciprocal of P. Thus $m = 1/P$.

The diametral pitch is now widely used, especially for machine-cut gears: calculations of pitch circle diameter do not involve the quantity $\pi$, and for many pitches this diameter (and hence the distance between shaft centres) is in round figures. **Metric module pitch**—the ratio, pitch diameter in *millimetres*/number of teeth—is almost exclusively used in Continental practice.

If D = pitch circle diameter, and T = number of teeth, then in symbols:

$$P = T/D \quad \text{or} \quad D = T/P, \quad . \quad . \quad . \quad (1)$$

$$m = D/T \quad \text{or} \quad D = T \times m. \quad . \quad . \quad (2)$$

The following relationships between the pitches should be noted:

$$\text{Circular pitch } p = \pi \,.\, D/T.$$

Hence

$$p \times P = \pi \quad . \quad . \quad . \quad (3)$$

or

$$p = \pi/P = \pi \times m. \quad . \quad . \quad (4)$$

**Tooth Proportions.**—For ***standard*** teeth the addendum is made equal to the module $m$, and the dedendum is made equal to the addendum plus clearance $c$.

The value of $c = p/20 = (\pi \times m)/20 = 0{\cdot}157m$ or $0{\cdot}157/P$.

For ***stub*** teeth the addendum and dedendum are reduced (see page 116).

**Pressure Angle** $\psi$ (*psi*).—The proportions for standard teeth are used in conjunction with pressure angles of either $14\frac{1}{2}°$ or $20°$: the latter angle is now favoured for reasons discussed later.

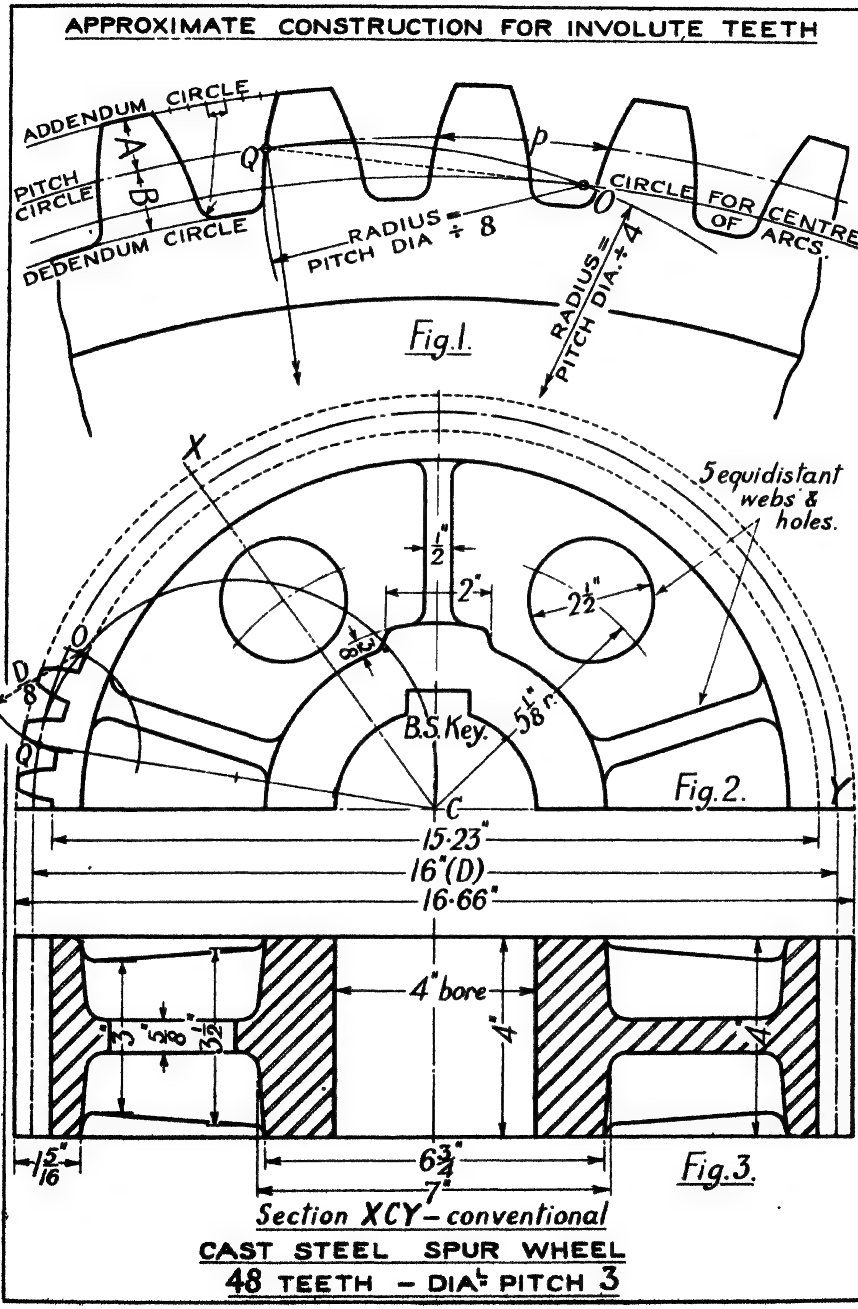
**To draw a Gear Wheel.**—Let the number of teeth T = 48; diametral pitch P = 3. From relationships (1) to (4) the following may be written down:—pitch diameter $D = 48 \div 3 = 16''$; module $m$ = addendum $A = \frac{1}{3}'' = 0{\cdot}3333''$; clearance $c = 0{\cdot}157 \div P = 0{\cdot}0523''$; dedendum $B = A + c = 0{\cdot}3857''$; whole depth $= A + B = 0{\cdot}719''$; circular pitch $p = \pi/P = 1{\cdot}047''$; outside diameter $= D + 2m = 16{\cdot}666''$; root diameter $= D - 2B = 15{\cdot}23''$. ***This is the gear shown opposite.***

**Approximate Construction for Involute Profiles.***—For gears of 30 teeth and more, the involute curve may be represented by a circular arc, radius D/8. The centres for the arcs are located as follows—refer to figs. 1 and 2. Draw the pitch circle and space off divisions = $\frac{1}{4}$ circular pitch. Join one point Q to the pitch-circle centre C, and upon QC as diameter describe a semicircle. With Q as centre, and radius D/8, describe an arc cutting the semicircle in O. This point O is one of the required centres. Evidently the other centres lie on a circle through O concentric with the pitch circle. The arcs are limited by the addendum and dedendum, and strengthening fillets are provided at the roots: radii of fillets = widest part of tooth space ÷ 7.

## EXERCISE

**Draw, half size, the following views of the cast-steel spur wheel shown in figs. 2 and 3: elevation of complete wheel omitting the teeth but inserting pitch, addendum and dedendum circles; sectional plan on XCY. Then draw, twice full size, four or five teeth as in fig. 1: use the approximate construction given above and confirm it by plotting the correct involute. Take $\psi = 14\frac{1}{2}°$.**

* Used by Messrs. Brown and Sharpe for involute teeth, pressure angle $14\frac{1}{2}°$.


APPROXIMATE CONSTRUCTION FOR INVOLUTE TEETH
ADDENDUM CIRCLE
PITCH CIRCLE
DEDENDUM CIRCLE
A
B
Q
p
O
CIRCLE FOR CENTRES OF ARCS.
RADIUS = PITCH DIA ÷ 8
RADIUS = PITCH DIA. ÷ 4
Fig. 1.
X
5 equidistant webs & holes.
1/2"
2"
2 1/2"
3/8
5 1/8" r.
B.S. Key.
D/8
O
Q
C
Y
Fig. 2.
15·23"
16"(D)
16·66"
4" bore
4"
3"
5/8"
3 1/2"
1 5/16"
6 3/4"
7"
Fig. 3.
Section XCY – conventional
CAST STEEL SPUR WHEEL
48 TEETH – DIA^L PITCH 3

**Pinion and Wheel.**—Involute gears will work correctly together if their teeth have the same pitch and the same pressure angle. Fig. 1 shows to scale a pinion A, of 30 teeth, and a wheel B, of 45 teeth, $P = 2$, $\psi = 14\frac{1}{2}°$. The teeth in contact are shown to a larger scale in fig. 2. (As already stated, the teeth cannot have involute contact beyond the points of tangency N and $N_1$. The length of the path of contact cannot be greater than the length of $NN_1$, and the *limiting* addenda circles are those which pass through N and $N_1$.)

The leading dimensions of the gears are obtained as on page 114:—Circular pitch = 1·5708″; addendum = 0·5″; clearance = 0·0785″; dedendum = 0·5785″; pitch diameters, wheel $22\frac{1}{2}$″; pinion 15″; centre distance = $18\frac{3}{4}$″. The gear ratio =

speed B/speed A = 30/45 = 2/3.

**Racks.**—A rack may be regarded as a wheel of infinitely large radius: hence the teeth have straight sides normal to the path of contact (i.e. inclined at $90° - \psi$ to the pitch line of the rack). Fig. 3 shows a 15-tooth pinion, $\psi = 20°$, in gear with a rack.

**Interference between Involute Teeth.**—When a pinion of few teeth engages with a large wheel, interference may occur between the faces of the wheel teeth and the flanks of the pinion teeth: interference occurs when a true involute pinion having less than 32 teeth ($\psi = 14\frac{1}{2}°$) engages with a rack.* Interference may be avoided by (*a*) shortening the addendum, (*b*) increasing $\psi$, (*c*) correcting the addendum.

A pressure angle of 20° is now widely used and is the standard recommended by the B.S.I. It gives a thicker root to the tooth and eliminates interference in many cases. In fig. 3 the faces of the rack teeth have practically no interference with the flanks of the pinion teeth. Complete freedom from interference in all cases is given by adopting either (*a*) or (*c*) together with (*b*).

In the D.B S. system † both (*b*) and (*c*) are used; the standard depth of tooth is retained, but the disposition of the addendum and dedendum relative to the pitch circle is altered. The pitch addendum and wheel dedendum are altered by equal amounts, in increments of 0·02 × module. In corrected addendum systems the width of tooth, measured on the pitch line, is no longer half the circular pitch, being greater for the pinion and less for the wheel.

In the Sellers and Logue systems the addendum and dedendum are reduced, giving what are known as **stub teeth.** Figs. 4, 5, and 6 show a comparison between Brown and Sharpe, Sellers, and Logue teeth for the same diametral pitch. In the Fellows stub-tooth system two pitches are used: the pitch diameter is reckoned from one pitch, and the tooth proportions from another.

**Advantages of Involute Teeth** over those of other shapes.—(1) The face and flank form one continuous curve (which may be readily ground); (2) the pressure angle is constant; (3) all gears having the same pitch and pressure angle will work together; (4) if the pitch circles do not exactly touch, the velocity ratio remains constant and the gears work equally well together.

## EXERCISES

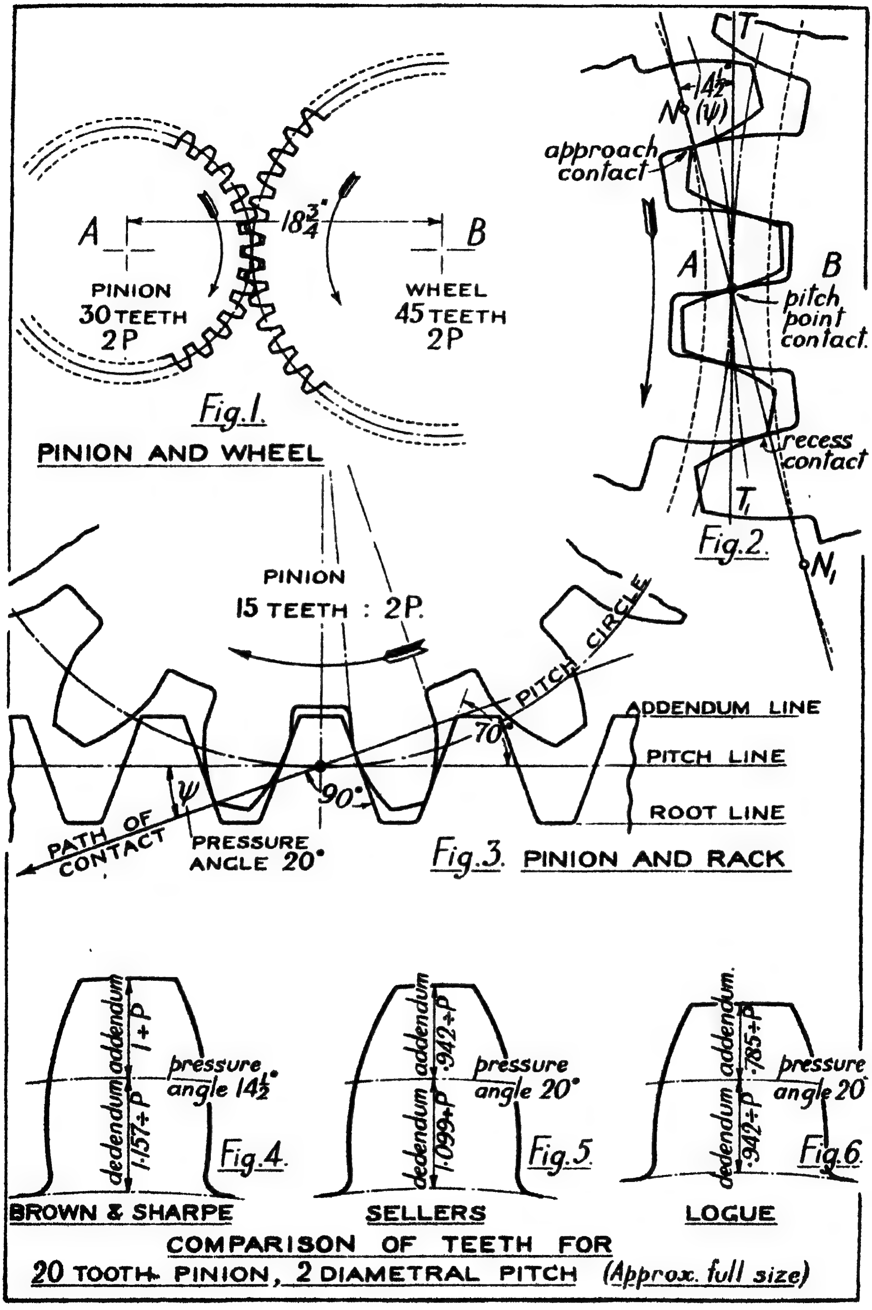
(1) Draw, full size, teeth profiles of the following gears showing a few of the teeth in mesh, as in fig. 2: (*a*) pinion 34 teeth, wheel 60 teeth, $P = 2$, $\psi = 14\frac{1}{2}°$; (*b*) pinion 17 teeth, wheel 30 teeth, $P = 2$, $\psi = 20°$.

(2) Draw, full size, about 6 teeth for an 18-tooth pinion, $P = 1\frac{1}{2}$, $\psi = 14\frac{1}{2}°$, and, on tracing-paper, a portion of a suitable rack (linear pitch 2·09″). Apply the rack over the pinion, and show that interference occurs.

(3) Repeat Exercise (2) with $\psi = 20°$ and show that interference is negligible.

* Hence teeth formed by rack cutters may have undercut flanks unless a correction is made.

† D. Brown & Sons, Huddersfield, England. The system is discussed in Dr. Merritt's paper "The Technique of Gear Design", *Proc. Inst. Automobile Engineers.*


A
B
18¾"
PINION
30 TEETH
2P
WHEEL
45 TEETH
2P
Fig.1.
PINION AND WHEEL
T
14½°
N (ψ)
approach contact
A
B
pitch point contact.
recess contact
T₁
Fig.2.
N₁
PINION
15 TEETH : 2P.
PITCH CIRCLE
ADDENDUM LINE
70°
PITCH LINE
90°
ψ
ROOT LINE
PATH OF CONTACT
PRESSURE ANGLE 20°
Fig.3. PINION AND RACK
addendum 1 ÷ P
dedendum 1·157 ÷ P
pressure angle 14½°
Fig.4.
BROWN & SHARPE
addendum ·942 ÷ P
dedendum 1·099 ÷ P
pressure angle 20°
Fig.5.
SELLERS
addendum. ·785 ÷ P
dedendum ·942 ÷ P
pressure angle 20°
Fig.6.
LOGUE
COMPARISON OF TEETH FOR
20 TOOTH PINION, 2 DIAMETRAL PITCH (Approx. full size)

**Strength of Gear Teeth.**—If the speed of a gear wheel and the power transmitted are known, the force exerted at the pitch circle—called the **tangent line load**—may be calculated: see (5) opposite. A standard formula, due to Lewis,* connecting stress, dimensions, and tangent line load, is based on the assumption that the load transmitted by a gear is taken by one tooth at its outer edge—an extreme condition that rarely obtains with machine-cut gears.

**Lewis Formula.**—Let F = tangent line load in lb., $f$ = permissible working stress at the root of the tooth in lb./in.$^2$, P = diametral pitch, $b$ = face breadth in inches, T = number of teeth in the gear.

Then

$$F = \pi \, . \, b \, . \, f \, . \, y \div P, \quad . \quad . \quad (1)$$

where the value of $y$, called the **tooth factor**, is given by the following:

For a pressure angle of $14\frac{1}{2}°$,

$$y = 0{\cdot}124 - (0{\cdot}648 \div T). \quad . \quad (2)$$

For a pressure angle of 20°,

$$y = 0{\cdot}154 - (0{\cdot}912 \div T). \quad . \quad (3)$$

The value of $f$ is reduced as the speed increases to allow for unknown impact loads, and it is taken as the product of the *static* stress $f_1$ (i.e. the stress at the elastic limit for the material) and a factor C known as the **speed coefficient.** In the Barth formula, C is based on the velocity V of the pitch line; if V is in ft./min.,

$$C = 600 \div (600 + V).$$

The modified Lewis formula then becomes:

$$F = \left\{\frac{600}{600 + V} \cdot \frac{\pi \, . \, b}{P}\right\} f_1 \, . \, y, \quad . \quad (4)$$

the portion within the bracket being common to both wheel and pinion. The **face breadth** $b$ varies from $3p$ to $4p$—i.e. from $3\pi/P$ to $4\pi/P$.

**Values usually taken for $f_1$, in lb./in.$^2$**, are: nickel chrome steel (100 tons/in.$^2$ U.T.S.), 55,000; nickel chrome steel (60 tons/in.$^2$ U.T.S.), 33,000; 0·5 per cent C. steel, 27,000; cast steel, 20,000; cast iron, 6000; phosphor bronze, 12,000; raw hide, 3000.

**The Power Transmitted** by a gear in ft. lb./min. is the product of the tangent line load F in lb. and the velocity V of the pitch circle in ft./min. Hence.

$$\textbf{H.P. transmitted} = F \, . \, V \div 33{,}000. \quad . \quad (5)$$

**Surface Stresses.**—The surface of a tooth may be destroyed by crushing or flaking if the compressive stress along the line of contact is excessive. The computation of this stress is, however, beyond the scope of this book.

**Worked Example.**—A steel pinion of 15 teeth, speed 400 revs./min., engages with a cast-steel wheel of 60 teeth: P = 2, $\psi$ = 20°, face breadth = $4\frac{1}{2}''$. Find the horse-power which may be safely transmitted by each gear.

Pitch diameters: pinion = $7\frac{1}{2}''$, wheel = 30″.

Pitch line velocity $V = \pi \, . \, 7\frac{1}{2} \, . \, 400 \div 12 = 785{\cdot}7$ ft./min.

Tooth factors:

$y$ (pinion) $= 0{\cdot}154 - (0{\cdot}912 \div 15) = 0{\cdot}093$;

$y$ (wheel) $= 0{\cdot}154 - (0{\cdot}912 \div 60) = 0{\cdot}139$.

Quantity within brackets in equation (4)

$$= \left\{\frac{600}{600 + 786} \cdot \frac{22}{7} \cdot \frac{4{\cdot}5}{2}\right\} = 3{\cdot}06.$$

Tangent line loads:

F (pinion) $= 3{\cdot}06 \times 27{,}000 \times 0{\cdot}093 = 7700$ lb.;

F (wheel) $= 3{\cdot}06 \times 20{,}000 \times 0{\cdot}139 = 8494$ lb.

Hence **h.p. pinion** $= 7700 \times 785{\cdot}7 \div 33{,}000 = \mathbf{183{\cdot}3}$;

**h.p. wheel** $= 8494 \times 785{\cdot}7 \div 33{,}000 = \mathbf{202{\cdot}2}$.

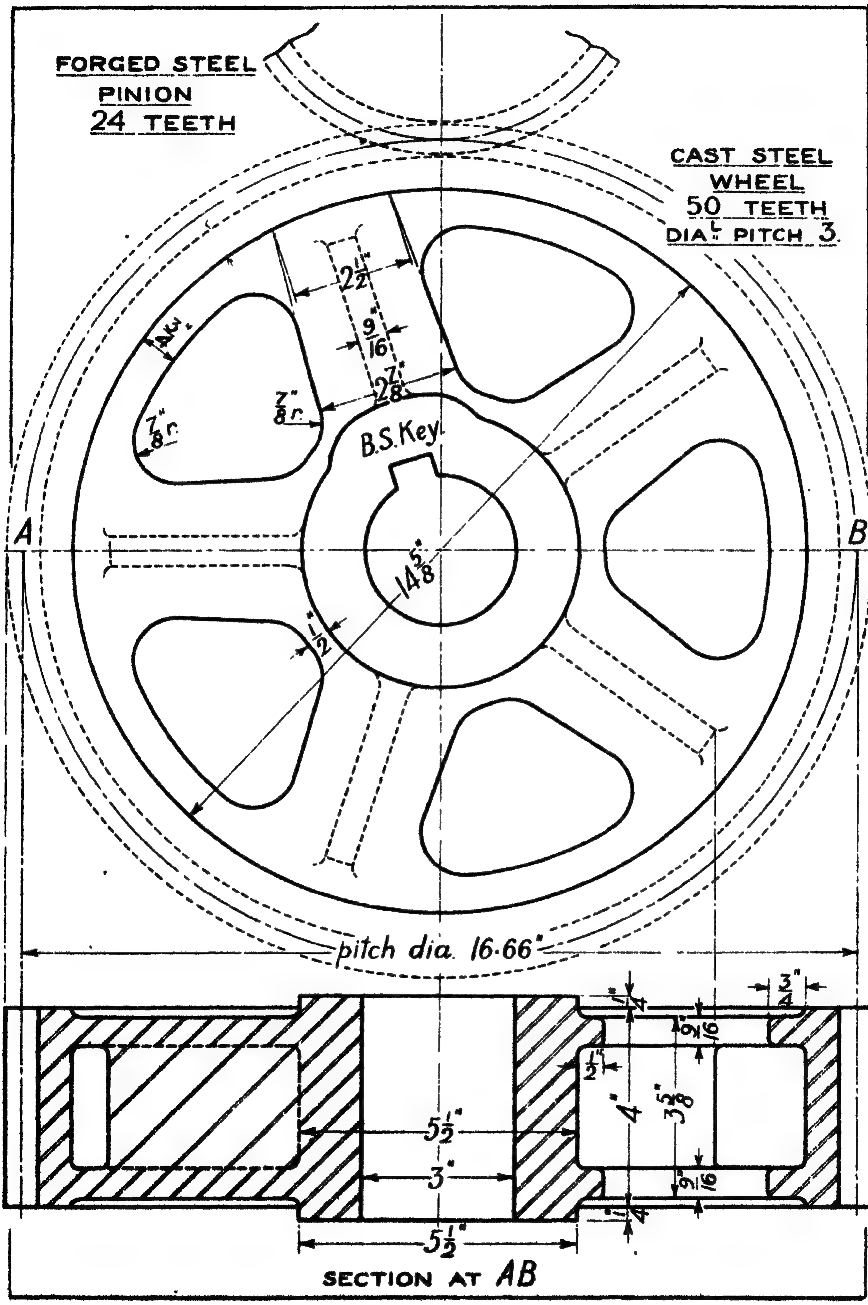
*Note.*—The above method of calculation has been largely superseded by the adoption of standard formulæ which take into account other factors—e.g. surface stress, fatigue, relative curvature: the student is referred to B.S.I. publication No. 436.

## EXERCISE

The gear shown is to transmit 60 h.p. at a speed of 60 revs./min. If the wheel is driven by a solid steel pinion of 24 teeth, 2″ bore, obtain the proportions of the pinion and make a working drawing of the two gears in mesh. Determine the assumed value of the static stress $f_1$ for wheel and pinion. Take $\psi$ = 20°.

*Answer.*—$f_1$, *wheel*, 19,100; $f_1$, *pinion*, 22,370 *lb./in.*$^2$.

* *Proc. Philadelphia Engineering Club*, 1893.


FORGED STEEL
PINION
24 TEETH
CAST STEEL
WHEEL
50 TEETH
DIAL. PITCH 3.
B.S. Key
A
B
pitch dia. 16·66"
SECTION AT AB

**Helical Gears** have been evolved from **stepped** gears, which consist of a number of identical spur wheels arranged side by side with the teeth slightly displaced relatively to each other, as in fig. 1. With stepped gears the phases of engagement overlap one another: when two teeth are in contact at the pitch line, other pairs are making approach and recess contact. In helical gears the teeth appear as a series of continuous helices, as in fig. 2, and the line of contact extends diagonally across the tooth face. As engagement occurs, pitch line contact starts at one end of a tooth and extends continuously across the tooth to the other end. Hence the load is never concentrated wholly at the outer edge of a tooth, as in straight gears, and the teeth are correspondingly stronger.

When a *single* helical gear is used, the obliquity of the teeth produces an axial thrust. This disadvantage is overcome by using **double helical gears,** as shown in figs. 4 and 6, in which equal gears of opposite " hand " are arranged side by side, either in one piece or separately. Double helical gears are widely used for the transmission of power at high speeds because of their smoothness of action and wearing qualities.

**Proportions.**—All sections of a helical gear taken perpendicular to the axis are similar: the involute tooth profiles are drawn as for spur gears. A pressure angle ($\psi$) of 20° is almost universal for helical gears. The teeth are usually of the stub type, common proportions being: addendum 0·88/P, dedendum 1·05/P. The spiral angle $\sigma$ (sigma), fig. 3, the angle between the helix and the axis, is frequently $22\frac{1}{2}°$ (although for marine turbine reduction gears 30° is used). For continuity of action the face breadth $b$ should be at least $= L/T$, where L = axial advance of a helix in one revolution, fig. 3, and T = number of teeth. This ratio $L/T = p \cot\sigma$, where $p$ = circular pitch; for $L = \pi D/\tan\sigma$, so that $L/T = (\pi D/T)$. $\cot\sigma = p \cot\sigma$. Hence when $\sigma = 22\frac{1}{2}°$, $b =$ or $> 2{\cdot}414\, p$; commonly $b = 2{\cdot}5p$.

**A Double Helical Pinion** of 15 teeth, 3P, is shown to scale in fig. 6. The half end view is drawn in the usual manner and the elevation projected from it. It is usually sufficient in the elevation to draw lines representing the tops and roots of the teeth, and to regard them as straight lines, not as helices. Here $L = \pi \,.\, 5/\tan 22\frac{1}{2}° = 37{\cdot}92''$. An axial movement of $2\frac{9}{16}''$ (the length of one gear) represents a rotation of $(2\frac{9}{16} \div 37{\cdot}92)360° = 24{\cdot}3°$. Hence the outlines of the same tooth (for the single gear) at its ends will appear in the positions A and $A_1$ respectively, fig. 5. The projection of the elevation of the tooth should be clear from fig. 5, and the complete elevation in fig. 6 involves a repetition of this for each tooth. There is, of course, no need actually to draw the dotted tooth profile.

If for any reason it is desired to plot the various helices accurately, points on the curves may be obtained by dividing up the axial and circular displacements, and projecting as shown by the dotted lines in fig. 5: the points lie very closely to the straight line $aa_1$.

**Correction for Interference.**—When a pinion has less than 15 teeth it is necessary to correct the tooth profile to avoid interference, as discussed on p. 116.

## EXERCISES

(1) Draw full-size views, as in fig. 6, of a double helical wheel of 30 teeth to engage with the pinion shown; the helices must be of the opposite hand.

(2) A double helical pinion for a shaft $2\frac{1}{4}''$ dia. has 21 teeth, 3P, spiral angle 30°. Obtain its proportions and prepare views corresponding to those in fig. 6.

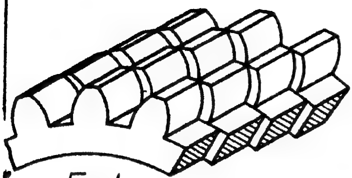


Fig. 1

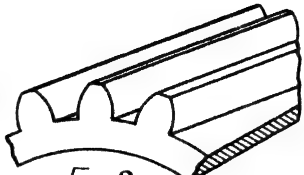


Fig. 2

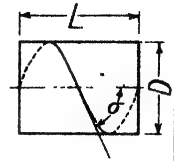



Fig. 3.

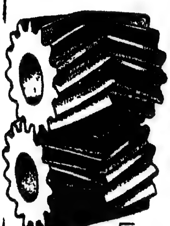


Fig 4

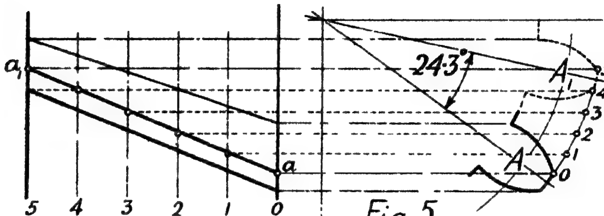
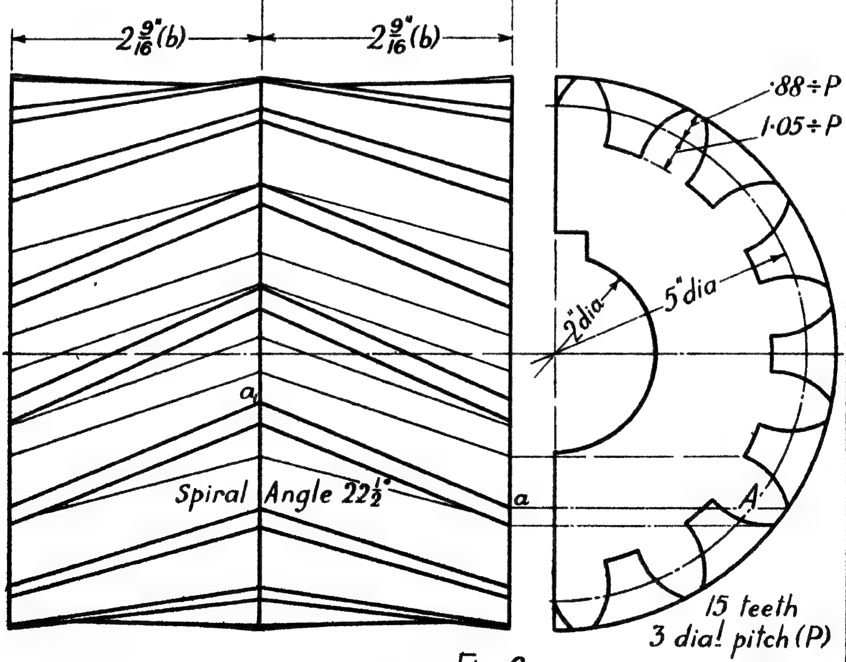



Fig 5.




ELEVATION Fig. 6. HALF END VIEW.

**Bevel Gearing** is used for the direct transmission of power between shafts whose axes intersect. The angle between the axes, $\Sigma$,* fig. 1, is commonly 90°, but it may have any value up to 180° (measured in the manner shown). The **pitch surfaces** are parts of cones having a common apex at the point of intersection of the shaft axes. The teeth may be supposed formed on these imaginary **pitch cones** by cutting grooves and adding strips between them, along straight lines radiating from the common apex. The base circle of the pitch cone is taken as the **pitch circle** of the gear. The angles $\theta_p$ and $\theta_w$, fig. 1, are respectively the **pitch angles** of the pinion and wheel. The **addendum** and **dedendum** are measured *at the large ends of the teeth*, and the **face width** is measured along a generator of the pitch cone.

The profiles of the teeth are set out on the surfaces of conical rims, shown shaded in fig. 2.† These **normal** or **back cones** have the same bases as the pitch cones, but corresponding generators are perpendicular to each other: e.g. $\angle ADC + \angle BDC = 90°$. In order to trace the teeth, the shaded surfaces must first be developed, i.e. laid out in a plane, and then, in effect, wrapped again round the respective cones with the teeth profiles marked upon them. The process is not difficult, for only the pitch circles need be developed (by drawing arcs, such as DE drawn from centre B with radius BD) and one or two teeth drawn upon them as for spur gearing.

**To draw a Bevel Wheel.**—*The student is advised to draw the example given as he reads the text.* The wheel shown, fig. 3, is one of two identical wheels for shafts at right angles, i.e. $\theta_p = \theta_w = 45°$: such wheels are known as **mitre wheels.** The wheel has 21 teeth, 2 diametral pitch; hence the diameter of the pitch circle is $10\frac{1}{2}''$.

Begin with the elevation. Draw CD, the pitch circle diameter, AO the axis, and the outlines CAD and CBD of the pitch and normal cones. With centre B and radius BD describe the arc DE, representing a part of the developed pitch circle. On DE draw the profiles of a few teeth ($P = 2$), exactly as on p. 115. The addendum and dedendum circles settle the face and root points, $f$ and $r$, and the complete tooth is given in section by joining $r$A and $f$A and marking off the face distance along DA.

Now advance the plan. With centre O draw the six circles representing the pitch, addendum, and dedendum circles for both ends of the teeth. Mark off radial centre lines for the teeth, and transfer the distances $a$, $b$, $c$ from a *developed* tooth, as shown for one tooth. Complete the plan as shown on the left: straight lines are radial from O and the curves are approximate arcs.

The completion of the elevation, shown on the left, is a simple matter of projection, using the plan and the sectional elevation.

The dimensions given are all that are essential, but usually either the face and root angles, $\phi$ and $\rho$, or the angles $\alpha$ and $\beta$ are marked on a working drawing. (Refer also to p. 185, on which are shown two unequal bevel gears.)

**The spur wheel of which DE is the pitch circle is known as the virtual spur gear; it is used in strength calculations on bevel wheels.**

EXERCISE

**Draw, half size, the following views of two mitre wheels in gear arranged as in fig. 1, each to the sizes given in fig. 3: elevation, showing the vertical wheel and half the horizontal wheel in section; a complete plan.** ***Calculate*** **$\phi$ and $\rho$.**

* The following Greek letters are used on this page: $\Sigma$ sigma, $\theta$ theta, $\phi$ phi, $\rho$ rho, $\alpha$ alpha, $\beta$ beta.

† Theoretically the profiles should be set out on spherical surfaces (ss—dotted—fig. 2), but as these are not capable of development conical surfaces are used. The approximation is due to Tredgold.

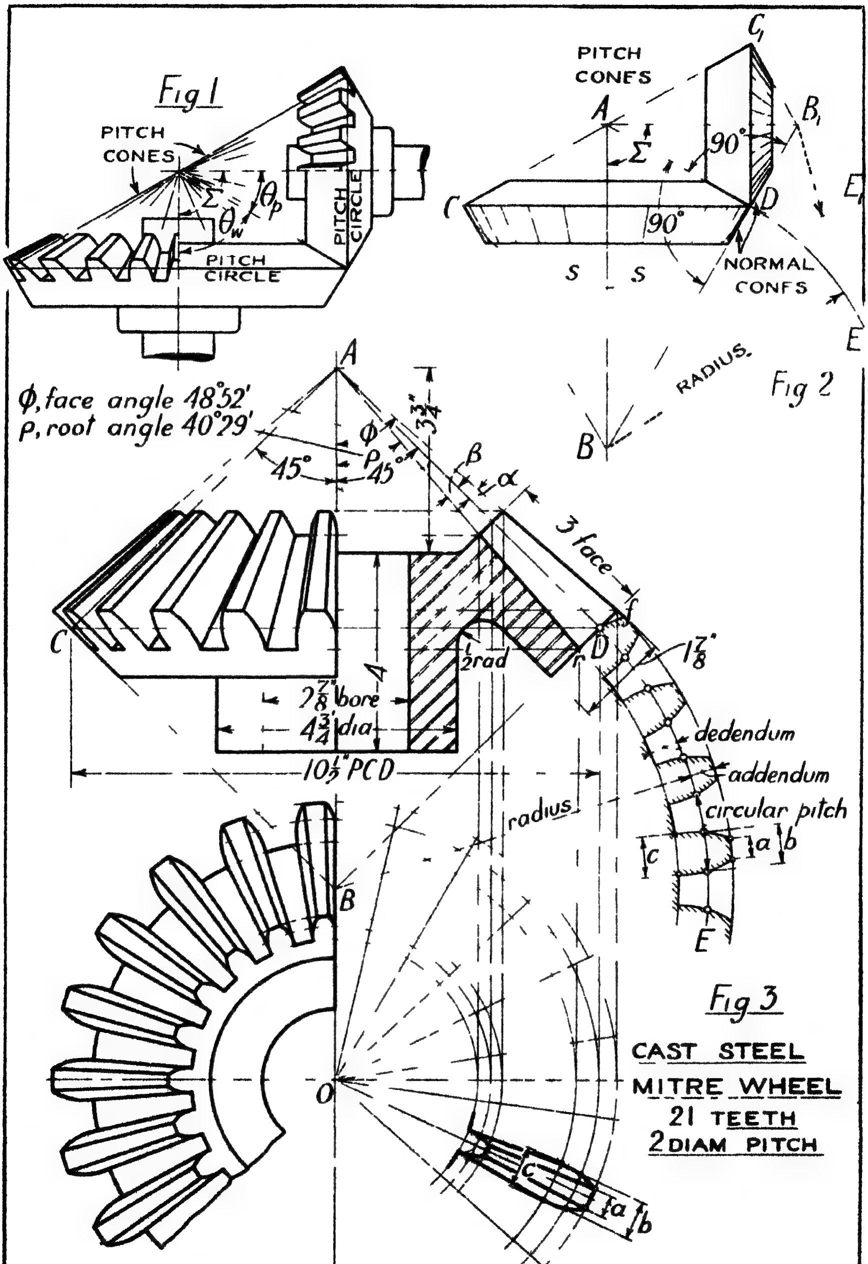


Fig 1
PITCH CONES
PITCH CIRCLE
PITCH CIRCLE
PITCH CONES
NORMAL CONES
RADIUS.
Fig 2
φ, face angle 48°52'
ρ, root angle 40°29'
45°
45°
3¾"
β
α
3 face
½ rad
2⅞" bore
4¾" dia
10½" PCD
1⅞"
dedendum
addendum
circular pitch
radius
Fig 3
CAST STEEL
MITRE WHEEL
21 TEETH
2 DIAM PITCH

**In worm gearing** the threads of a revolving screw press against teeth cut on the face of a wheel, fig. 3, and cause it to rotate: the axes of the worm and wheel are usually at right angles. The ordinary type of worm is straight-sided on an axial section, and the thread proportions are those of a standard rack tooth, as shown in fig. 4: on a central section the wheel teeth are of involute form.

Refer to figs. 1 and 2. The lead L of a worm is the axial advance of a thread per revolution. The lead angle $\lambda$ (lambda) is the inclination of the pitch helix to a plane perpendicular to the worm axis. Hence $\tan\lambda = L \div \pi \, . \, d$, where $d$ is the pitch diameter of the worm. For high efficiency $d$ should be small and $\lambda$ large: maximum efficiency is given when $\lambda = 45°$ approx. When $\lambda$ exceeds 9°, however, the drive is reversible: hence irreversible gears are relatively inefficient.

**Conventional Representation of Worm Gearing.**—The accurate projection of the wheel teeth is a laborious operation, and it is sufficient to show a central section of part of the wheel, and to indicate the pitch and other circles by broken lines, as opposite. The *true* section of the worm in the end views is not plotted. Frequently, in practice, the worm and wheel are shown as blanks, as on pp. 196 and 197.

In fig. 4 the helical outlines of the worm threads have been plotted for the lower half, and the construction for one helix is shown. In fig. 5 straight lines have been used for the worm threads.

**Proportions, Materials, &c.** Refer to fig. 4.—The gear shown has a velocity-ratio (R) of 20, and centre distance (C) of 12″. Values $\lambda = 17°$ and $t = 2$ have been chosen arbitrarily. The remaining proportions are arrived at thus: teeth in wheel, $T = R \, . \, t = 40$; pitch diameter of worm, $d = 2C \div (1 + R \tan\lambda) = 3{\cdot}38''$; pitch diameter of wheel, $D = 2C - d = 20{\cdot}62''$; pitch of teeth, $p = \pi D \div T = 1{\cdot}62''$; addenda, $A = a = 0{\cdot}3183p = 0{\cdot}51''$; dedenda, $B = b = 0{\cdot}3683p = 0{\cdot}59''$;* overall diameter of wheel $= D + 3A = 22{\cdot}15''$.

The worm is usually of steel, case-hardened and ground. The wheel rim should be of G.M. or Phos.B., mounted on a C.I. centre. A suitable housing for the gear in fig. 5 is shown on pp. 196 and 197. The axial thrust on the worm is usually taken by ball thrust bearings, but if the thrust is small it may be carried by the radial type of ball bearing as on p. 196. It should be noted that the shaft is unnecessarily weakened by reducing the diameter at the ends, as in fig. 5: the form indicated on p. 196 is preferable.

**Interference.**—When $\lambda$ exceeds 20° interference may occur on the leaving side of the rim if a small pressure angle, e.g. 14½°, is used: for this, and for other manufacturing reasons, larger pressure angles (up to 30°) are adopted. As in spur gearing, undercutting may be avoided by adjusting the addenda and dedenda; the example shown in fig. 5 illustrates the effect of this adjustment.

**Special Worm Forms.**† — Patented worm gears giving high efficiencies are now largely used. The worms are sometimes shaped like an hour-glass to fit the wheel and are known as globoidal, encircling, or Hindley-type worms. In all good designs, the line of contact should be approx. radial, so that a lubricating film may be maintained between the threads and wheel teeth.

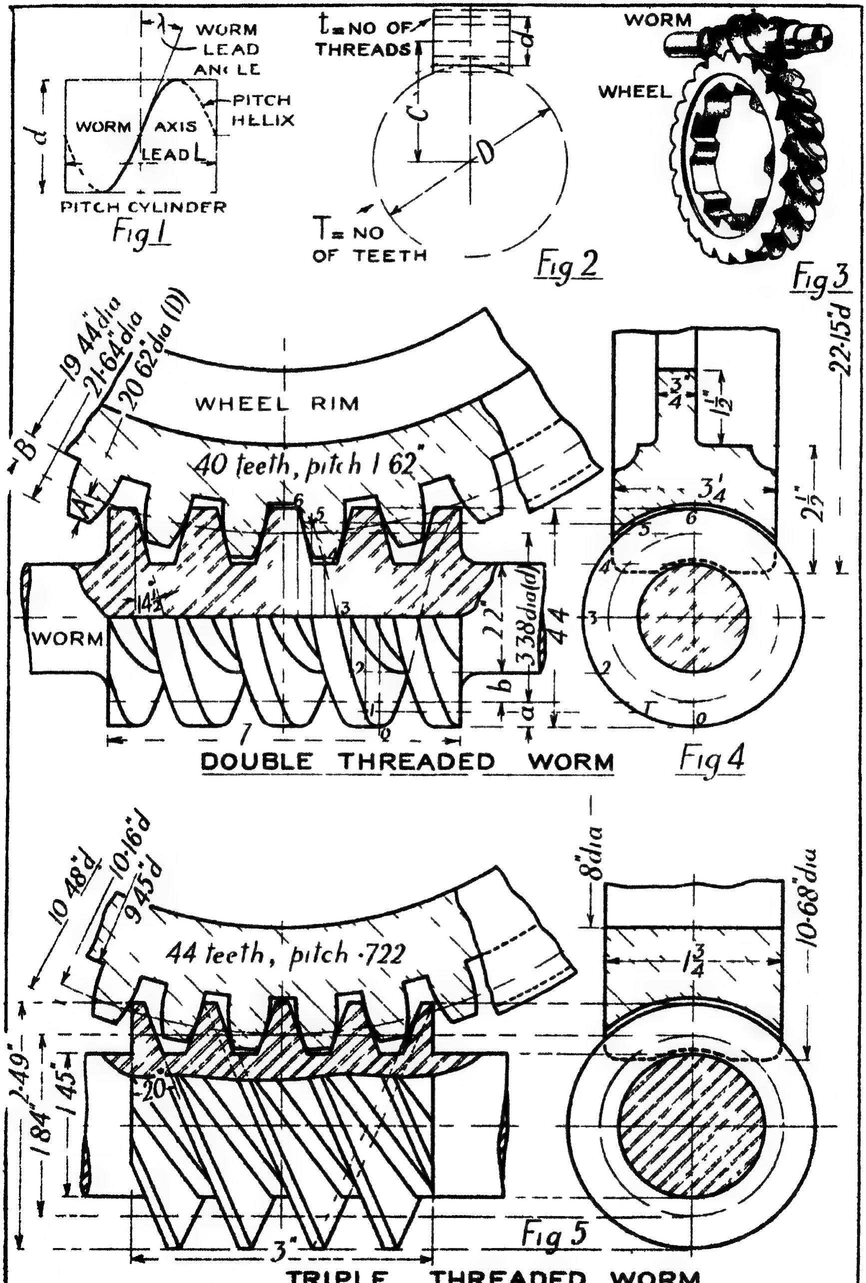
## EXERCISES

(1) Set out, half size, views, in the manner shown in fig. 4, of the given double-threaded worm and wheel.

(2) Draw, full size, views, in the manner shown in fig. 5, of the following worm and wheel. Worm: triple thread, right-handed, pitch dia. 2½″, lead 2¾″, pressure angle 20°, length 4½″. Wheel rim: 45 teeth, width 2″, least thickness 1″.

* For triple- and quadruple-threaded worms it is the practice in U.S.A. to make $A = a = {\cdot}286p$ and $B = b = {\cdot}332p$.

† *Proc. I. Mech. E.*, 1936, "Worm Gear Contacts", Abbott.


WORM LEAD ANGLE
PITCH HELIX
WORM AXIS
LEAD L
PITCH CYLINDER
Fig 1
t = NO OF THREADS
T = NO OF TEETH
Fig 2
WORM
WHEEL
Fig 3
WHEEL RIM
40 teeth, pitch 1 62"
19 44" dia
21·64" dia
20 62" dia (D)
WORM
3 38 dia (d)
22·15" d
DOUBLE THREADED WORM
Fig 4
44 teeth, pitch ·722
10 48" d
10·16" d
9·45" d
8" dia
10·68" dia
2·49"
1 84"
1 45"
3"
Fig 5
TRIPLE THREADED WORM

**Valves** are used to check or control the flow of fluids through pipes or cylinders. They may be divided into three classes: (*a*) valves which rotate in opening, e.g. flap and throttle valves; (*b*) valves which rise and fall perpendicularly to their seats, e.g. lift valves for pumps; (*c*) valves which slide over their seatings, e.g. sluice valves, engine slide valves. Each class may be subdivided into types that are operated by the action of the fluid, and types that are controlled by hand or by some mechanism. Valves and their casings have many forms and only a few representative examples can be discussed here.

In all valves the design must be such that they are tight when closed and offer the minimum resistance to the flow of fluid when open. Automatic valves should close promptly; they are often assisted to do so by springs. The clear area through the valve seating is determined by assuming a limiting speed for the fluid; for water, a velocity of 3 ft./sec. is usual. If Q is the quantity flowing through the valve in ft.$^3$/sec., and V is the velocity in ft./sec., then the area A in ft.$^2$ is given by $A = Q/V$.

**Low-pressure Flap Valve.**—The drawing shows, in section, a simple flap valve and its C.I. casing or box, suitable for a working pressure of 50 lb./in.$^2$. The valve is operated by the fluid and permits flow in one direction only, from left to right: a reversal closes the valve.

The seating consists of a projecting annulus cast around one of the branch openings and accessible for machining through the opening at the top of the casing. For the design shown, the casing should be arranged with the branches horizontal, so that the unloaded valve bears with a slight pressure on the seat, which is inclined to the vertical. The flap, shown separately to a larger scale, is a C.I. disc, stiffened by means of ribs, and faced with leather. The leather disc is secured to the flap by an iron ring and four iron bolts, the ends of the bolts being slightly riveted over to prevent the nuts from working loose. The projecting pins at the top of the disc drop into supports cast on the side of the box, ample clearance being provided around the pins.

A flat supporting surface is usually cast on the underside of the casing: this has been omitted here to give a simplified design. For services where it is vital that the valve should function properly, e.g. the water-cooling system of an oil-engine, glass observation panels are often provided at the side of the valve box.

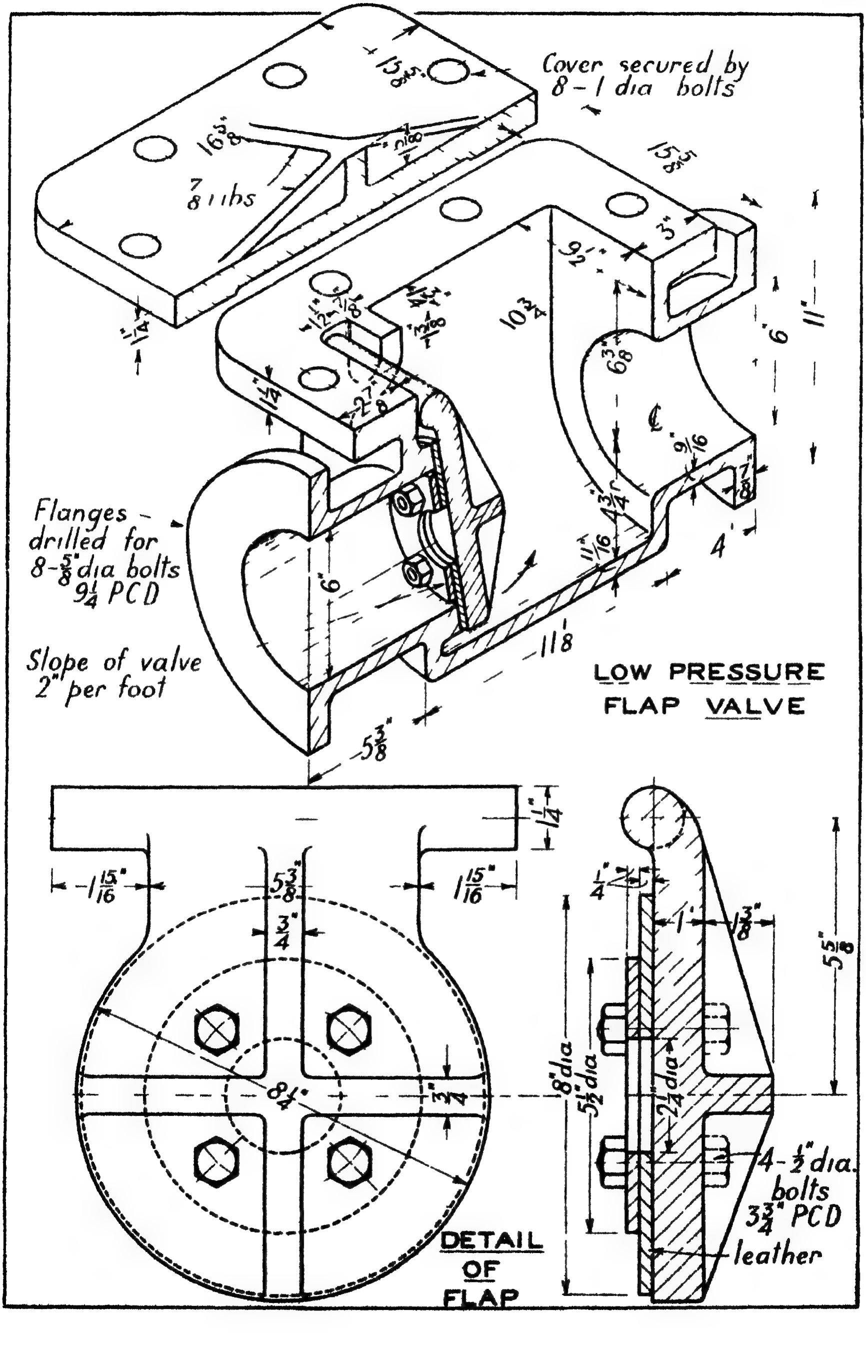
This type of valve is cheap, durable, and requires little attention in use.

**Butterfly Valves** are formed by placing two flap valves edge to edge over adjacent seatings. Usually the flaps are arranged horizontally.

## EXERCISES

(1) Draw, half size, the following views of the flap valve, showing the cover bolted in position: elevation, with the left-hand portion in section to reveal the valve; plan, showing the front half of the cover removed; sectional end view on valve face with valve removed. Insert all necessary dimensions and indicate those surfaces that require machining. Use your own judgment in determining the exact positions of the cover bolts and in settling minor dimensions.

(2) Prepare working drawings of a flap valve similar to that shown but suitable for a $3\frac{1}{2}''$ bore pipe line.


Cover secured by 8 - 1 dia bolts
7/8 ribs
Flanges drilled for 8-5/8" dia bolts 9 1/4 PCD
Slope of valve 2" per foot
LOW PRESSURE FLAP VALVE
DETAIL OF FLAP
4-1/2" dia. bolts 3 3/4" PCD
leather

**Lift Valves for Pumps.**—The drawing shows a common type of pump valve consisting of three thin discs, of diminishing diameter upwards. The discs are located by a central stud screwed into the valve seat, and their lift is limited by a guard secured to the stud. A light spring, housed within the guard, presses on the discs and ensures their prompt return to the seating during reflux. The seating is a ribbed casting, ground on its upper or working face, and threaded externally to suit a screwed hole in the pump chamber. Usually the screwed parts are tapered to ensure tightness. The nut is sawn partly through and closed slightly by hammering before use to prevent its slackening back under vibration. The spring and discs are of phosphor bronze; all other parts are of gunmetal.

**Non-return Lift Valve and Box.**—Another type of lift valve is shown with its box or casing in the figure opposite. The valve is arranged with its axis vertical so that its weight always tends to close it. The seating is a G.M. bush pressed into the C.I. casing and rolled into the dovetailed recess before being finally machined.* The valve is guided in the bush by three ribs or feathers undercut at the top as shown in the pictorial view, and its lift is limited by a projection from the cover. The ribs are sometimes given a slight twist, so that each surge of fluid past the valve turns it and compels it to seat itself in a new position, thus making the wear uniform. The face of the valve and of the valve seating are inclined at 45° to the axis. The width of the seating varies from $\frac{1}{16}''$ in small valves to $\frac{3}{16}''$ in large valves: the area of the seating should be such that the maximum load on the valve does not produce a greater crushing stress than 2000 lb./in.$^2$ (for G.M.).

**Feed Check Valves** for boilers are commonly of the type shown, but with these a screwed spindle is usually provided in the cover for regulating the lift of the valve or closing it altogether: they are then called screw-down non-return valves. Boiler-feed valves are usually of gunmetal or cast steel.

**Screw-down Stop Valve and Box.**—An example of this type of valve is shown on pp. 180 and 181. The valve is lifted by a screwed spindle, the enlarged end of which enters a slot in the valve. The method of guiding the valve should be noted.

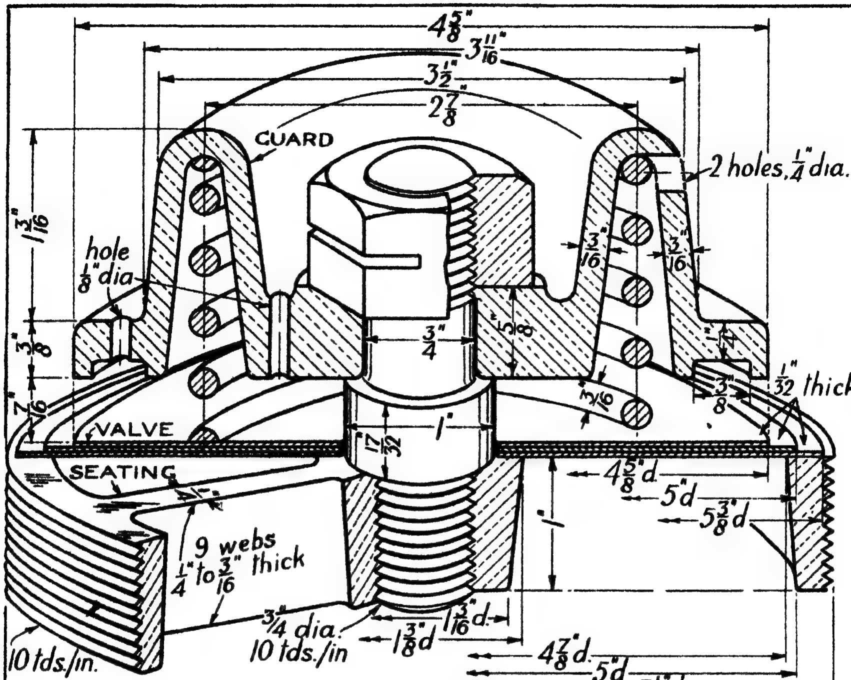
**Lift of Valves.**—If D = diameter of valve seating, and L = lift of valve, the area of the waterway around the circumference = $\pi . D . L$. For this to equal the area through the seating ($\frac{1}{4} . \pi . D^2$) the lift L must equal $\frac{1}{4}$D. Evidently the lift may be reduced by arranging the seating to accommodate more than one valve.

## EXERCISES

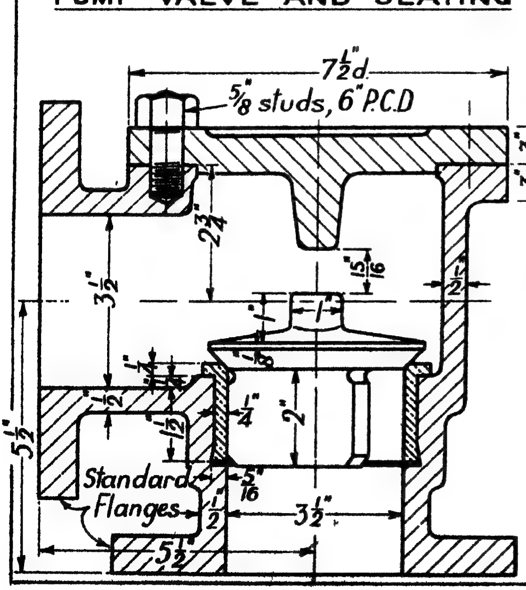
(1) Draw, full size, the following views of the pump valve shown: elevation, half in section; plan, one half on the seating. Dimension the views. Calculate the clear area through the seating and compare it with the waterway when the discs are touching the guard.

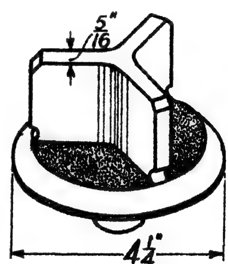
(2) Draw, half size, the following views of the complete valve and box shown: elevation, with the left-hand portion in section (do not section the valve); end view from left to right; plan with cover removed. Insert all necessary dimensions and machining marks. Note: certain details (e.g. number of cover studs) and dimensions have been left for the student to settle.

* As an alternative the bush may be secured by a screw, entered from the outside.




PUMP VALVE AND SEATING




Pictorial View on underneath of Valve

VALVE AND BOX

NON-RETURN TYPE

WORKING PRESSURE

100 lb/in.²

**Slide Valves.**—The drawings show a simple type of valve used to control the flow of steam to the cylinder of an engine. The valve is of D section. The flanged face of the valve slides over a plane seating on the cylinder, in which are cut three rectangular ports or openings. The outer ports lead to the ends of the cylinder, and the middle port leads either to the atmosphere or to a condenser. When the valve is placed centrally over the ports it covers them completely (usually), as in fig. 1. Movement of the valve from the central position permits steam to pass through the uncovered opening to one end of the cylinder. The valve is made to reciprocate by gear actuated by the engine crank, and its motions and proportions are so arranged that whilst steam is passing through one outer port to one side of the piston in the cylinder, steam from the other side is returning through the other outer port and is passing to the middle or exhaust port via the cavity in the valve. Fig. 2 shows the valve in one of its extreme positions.*

The amount by which the valve overlaps the steam ports in its mid position is called the **lap**: L, fig. 1, is the "steam" lap and *l* the "exhaust" lap. The latter may be negative, i.e. the edge of the valve may fall short of the edge of the port. The amount of opening to steam when the piston is at the end of its stroke is called the **lead**.

**The Locomotive Slide Valve** shown partly sectioned in fig. 3 is made of high-grade C.I. and is machined on the working face and its edges, and at the ends of the trough. The valve is located between two steel yoke pieces: these bear on the machined sides of the valve and are screwed on to the valve rod. Projections on the yoke pieces fit into the machined ends of the trough in the valve. The valve is free to move transversely between the yoke pieces and to take a close bearing on the seat under the pressure exerted by the steam on its back.

The port areas are settled by allowing steam speeds varying from 80 to 110 ft./sec. The length of each port is limited to about the diameter of the cylinder.

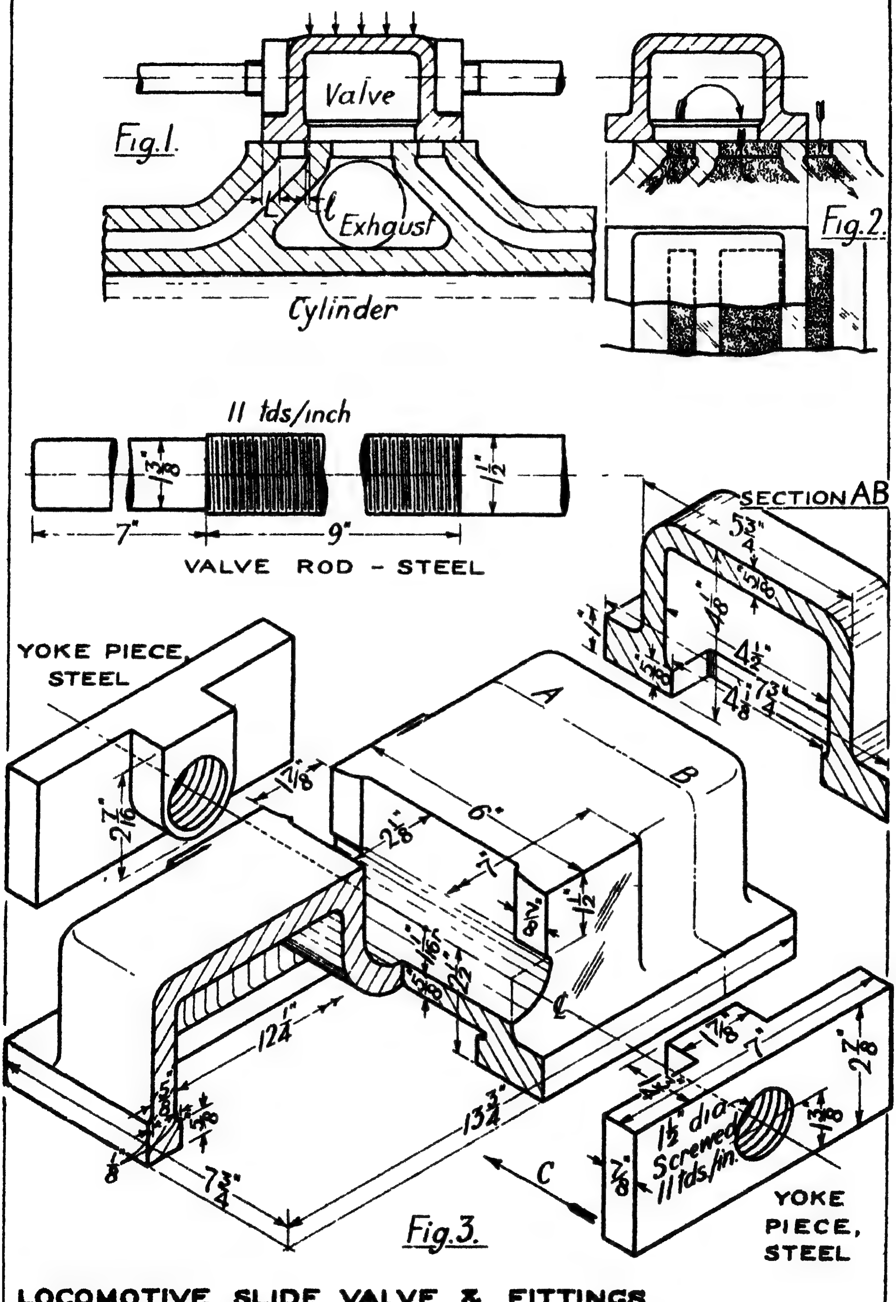
The frictional resistance to the motion of a large valve pressed on its seating by high-pressure steam may be considerable, and for this and other reasons valves of the piston type are usually fitted to H.P. cylinders. A **piston valve** is shown on page 187, this being suitable for the cylinder shown on page 147.

**A Locomotive Regulating Valve** of the sliding type is shown on pages 182 and 183. It is fixed in the dome of the boiler and controls the steam supply to the engine. Two valves are used, one sliding over the other. The outer or "pilot" valve is relieved at the back and is easily moved under steam pressure. A small displacement of the pilot valve permits steam to pass to the pipe through a small port in both valves: this steam exerts a pressure on the inner face of the main valve, thus partly relieving the load upon it and rendering its movement relatively easy. The drawings will repay a careful study and the student is advised to cut the sections out in cardboard.

## EXERCISE

Draw, half size, the following views of the slide valve shown, with yoke pieces and rod (broken) in position: elevation in direction of arrow C, half in section; end view, half in section on the rod axis; plan; underneath view. Insert machine marks and all necessary dimensions.

* The invention of this type of valve is credited to Murdoch, James Watt's assistant.




**LOCOMOTIVE SLIDE VALVE & FITTINGS.**

**A piston** is a cylindrical piece which reciprocates in a hollow cylinder, being moved by the fluid entering the cylinder. A **piston rod** connects the piston to mechanism outside the cylinder. When the fluid acts on each side of the piston leakage is prevented, where the rod passes through the cylinder end, by means of a stuffing box or gland (see page 139). In single-acting engines the piston rod may be dispensed with and a **trunk piston** used. In simple pumps an enlarged rod or **plunger** is used in lieu of a piston. Pump pistons are commonly called **buckets.**

Leakage of fluid past the piston may be prevented by fitting split metal rings into grooves in the piston. Each ring is machined to a diameter slightly larger than the cylinder bore, and a gap is cut across it; the ends are then drawn together and the outside diameter machined to the exact bore. When the rings are sprung into position they exert a constant pressure on the cylinder walls.

**Pistons for Locomotives** were at one time invariably made of C.I. They are now often made of steel, either cast or pressed, to reduce weight. Fig. 1 shows a C.I. piston of the single-disc type, suitable for a moderate pressure and speed. The rings are of C.I. and are located by pegs or dowels to prevent the joints from working into line. The piston boss is bored to fit a coned end on the piston rod and is secured by a cottered nut.* The taper on the rod varies from 1 in 6 to 1 in 8, on the diameter:† a smaller taper renders difficult the removal of the piston, and a larger taper unduly reduces the diameter of the rod. The complicated form of the piston prevents any exact theoretical estimation of its strength, and empirical rules are largely used in design work.

**Pistons for Oil-engines** are of various types. That shown in fig. 2 is suitable for a marine Diesel engine, 4-stroke crosshead type, stroke 38″, developing 200 h.p. per cylinder when running at 125 revs./min. The piston rod terminates in a flange to which the piston is secured by steel studs. These studs are proportioned to take the maximum inertia load of the piston, the tensile load which they carry during the suction stroke being difficult to estimate.

The high temperature range to which the piston is subjected makes its design a problem mainly for the metallurgist. The modern tendency is to eliminate strengthening ribs and to allow the fullest possible freedom for expansion and contraction (see also remarks on page 136). Both piston and rings are of special C.I. The piston is water-cooled, but details of the gear have been omitted in the sketch. Piston clearances (measured on the diameter) are 0·1″ at the top and 0·02″ at the bottom. Eight rings are provided, each cut diagonally; the upper two are located in position.

**A Trunk Piston** for a Diesel engine is described on page 136.

## EXERCISES

(1) Draw, half size, the following views of the piston in fig. 1 with the rod end in position, and dimension them: elevation, half in section; end view (on the nut). Devise a dowel arrangement for locating the rings. If the unbalanced pressure on the piston is 150 lb./in.$^2$, what is the direct stress at the root of the threads?

*Answer.*—*7690 lb./in.$^2$.*

(2) Draw, quarter size, the following views of the Diesel engine piston, and dimension them: elevation, half in section; underneath view (i.e. on the rod), one half showing the rod removed. Settle the number of studs by assuming a maximum inertia load of 15,000 lb. and using a stress value of about 4000 lb./in.$^2$.

* American practice is to rivet over the rod and, thereby dispensing with cotter and projecting end.

† James Watt used a taper of 1 in 5.

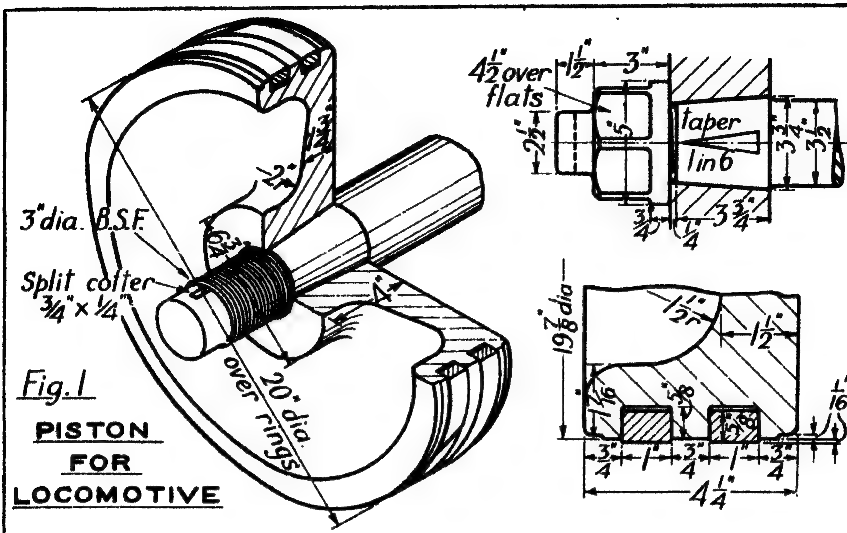



Fig. 1

PISTON FOR LOCOMOTIVE

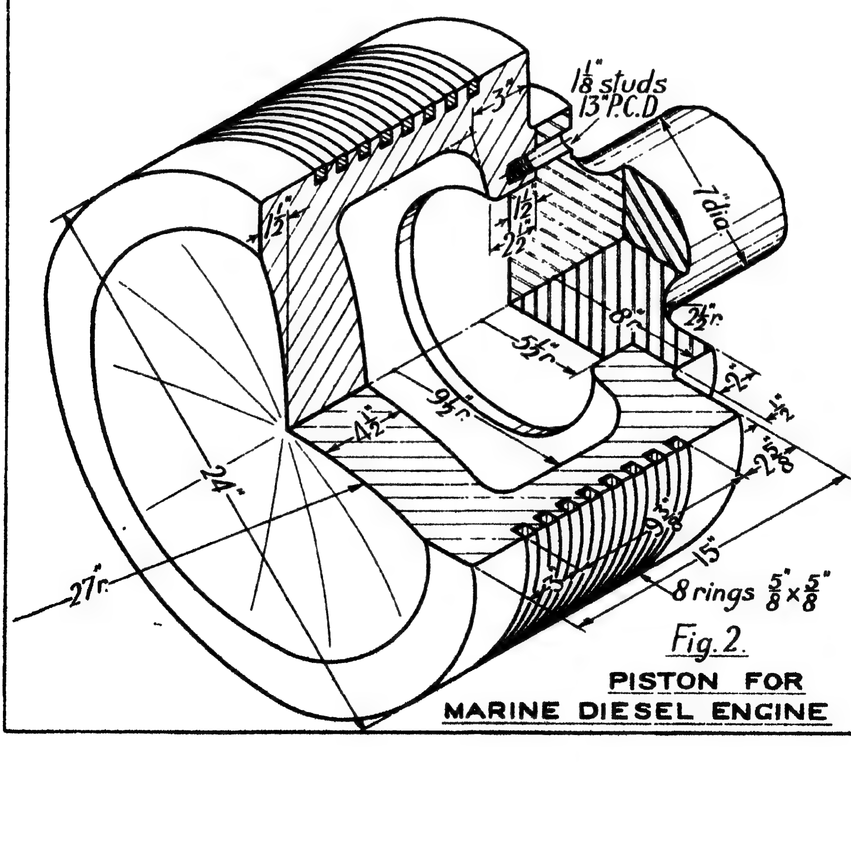



Fig. 2.

PISTON FOR MARINE DIESEL ENGINE

**Pistons for Marine Engines** are usually of steel (cast or forged) and conical in form: strength and rigidity together with lightness are aimed at. In a compound or triple-expansion engine the cylinders are of the same height and the piston rods of the same length: hence pistons for the one engine are made equally deep, both overall and at the rim. The depth is settled for the L.P. cylinder, allowing a slope at the underside of the piston of about 1 in 3. The piston rods are usually identical in size: the piston bosses, therefore, are the same in bore and depth.

The spring rings of the **H.P. piston,** fig. 1, are usually of the "restricted" type, of C.I. or Phos.B., fitted one on each side of a solid C.I. restraining ring. The latter limits the outward movement of the spring rings; when they become slightly worn they fit the cylinder closely and are prevented from expanding further by the shoulders on the restraining ring. High-pressure steam acting behind unrestricted rings would exert an undesirable force on the cylinder walls. Above the rings, secured to the piston by steel studs, is the junk ring,* either of steel or C.I. Thus arranged, the rings may be replaced without disturbing the piston.

The **L.P. piston,** fig. 2, carries one broad ring only, of C.I., kept pressed against the cylinder walls by springs; the springs are housed in pockets formed in the piston body and press radially on the ring.

Many spring arrangements have been evolved with the object of relieving the piston of any reaction, so that the ring may "float" in its groove, but space will not permit of a discussion of these.

Leakage past the opening in the ring is prevented by a tongue-piece, fitted as shown in the small figure.

The piston-rod ends are coned and provided with collars (or shoulders). The piston bosses are bored to fit the cones, but at the same time are kept just clear of the collars. The piston-rod nut, and the junk ring studs and nuts, must all be prevented from working loose. In the figure the former is locked by a thin steel plate, in turn secured to the piston by square-necked studs with castle nuts. The junk ring studs have square necks and castle nuts. Tapped holes must be made for lifting the piston and rings. Gear for forcing the rod from the boss is shown in the small sketch.

**Proportions.**—As the result of long experience, the principal dimensions of pistons of this type are practically standardized and are detailed in many marine engineering pocket-books.† The depth at the rim is commonly made equal to the diameter of the rod (for which see page 138). The thickness of the coned portion of the piston is given by the following formulæ (for C.S.):

$$\text{thickness at rim} = 0{\cdot}003D\sqrt{p} + 0{\cdot}15'',$$

$$\text{thickness at boss} = 0{\cdot}005D\sqrt{p} + 0{\cdot}26'',$$

where $D$ = cylinder diameter in inches, $p$ = pressure on piston in lb./in.$^2$. If $P$ is the boiler pressure, $p$ may be taken as $\cdot5P$ for H.P., $\cdot25P$ for I.P., and $\cdot17P$ for L.P.

**Hollow Pistons** are largely used for stationary engines, being stronger than the single-disc type: an example is given on p. **186**. The core supports leave holes in the base of the piston, and these are afterwards screwed and plugged.

## EXERCISES

Show each piston complete with fittings and with the rod end in position. Insert all necessary dimensions: some of these are purposely omitted in the drawings given.

(1) *H.P. Piston.*—Draw, half size, an elevation, half in section, and a half plan with a portion of the junk ring removed. Take dimensions of the boss from fig. 2. Show all necessary lifting holes.

(2) *L.P. Piston.*—Draw, quarter size, a half-sectional elevation and a part plan: in the latter show a portion of the junk ring removed to reveal the springs. Show separately, to a larger scale, a suitable joint in the ring.

* Pistons were originally packed around with junk (hemp and tallow) kept in place by a junk ring: the name has persisted.

† Refer to *Seaton and Rounthwaite's Pocket Book.*

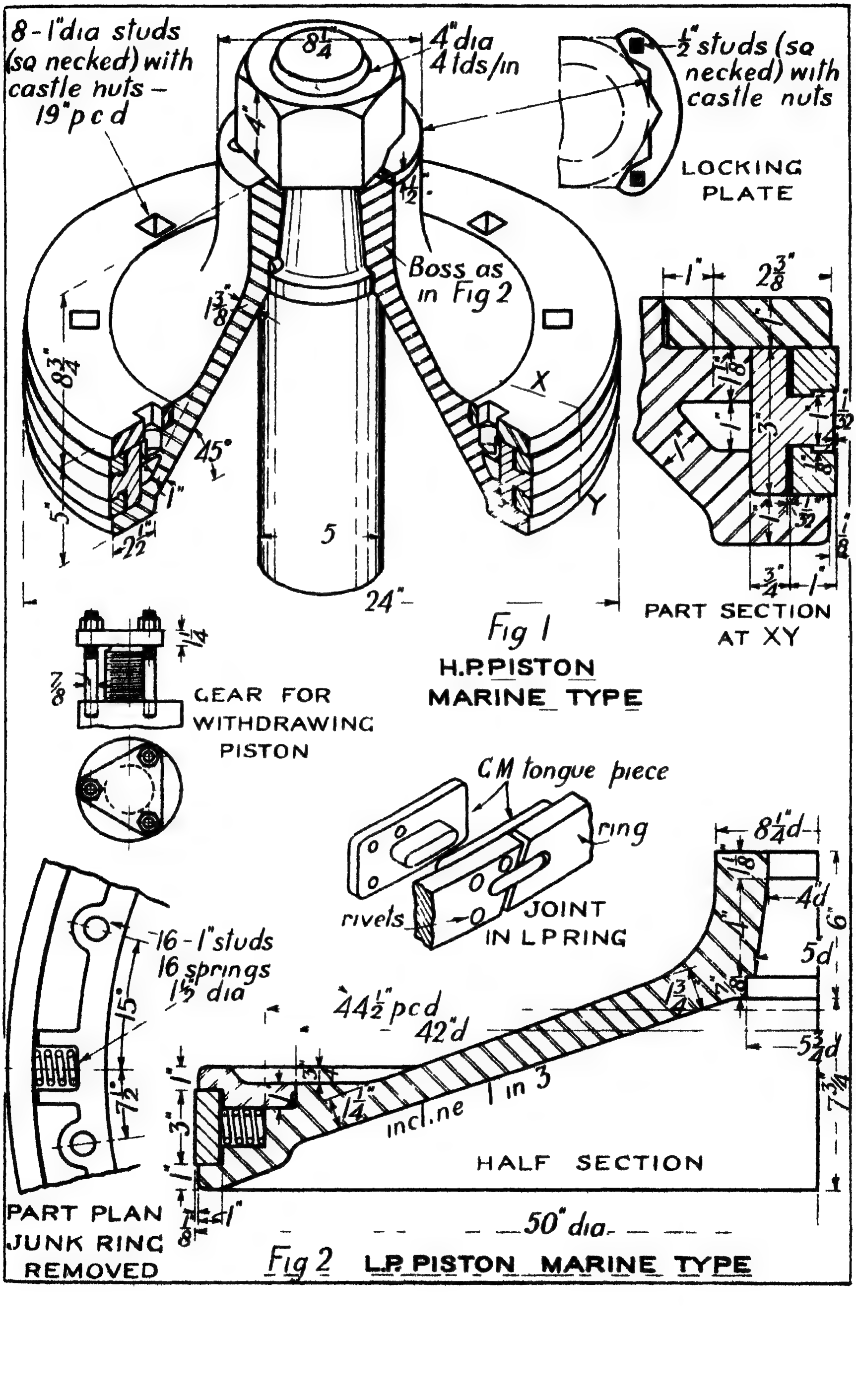



Fig 1 H.P. PISTON MARINE TYPE

Fig 2 L.P. PISTON MARINE TYPE

**Trunk Piston for Diesel Engine.**—Here piston rods and crossheads are dispensed with. Conditions of service for all Diesel engine pistons are severe. The large quantities of heat transmitted cause high temperature stresses in the metal, and the construction of a suitable piston is a problem for the metallurgist rather than the designer. The same remark applies equally to liners and cylinder heads. Cast iron is generally used for these parts, special irons having been produced to withstand the high temperatures without cracking. Aluminium alloys are, however, being successfully used for pistons, and a design is given on p. 201.

Any abrupt variation in the surface contour of a casting results in a loose arrangement of crystals, which group themselves along lines at right angles to the surface. Repeated heating tends to break down the cohesion of these loose crystals and cause cracking. Hence in Diesel engine pistons it is essential to avoid all sharp angles in the design as cast. Any necessary ribs or webs should join the main casting in fillets of large radius.

In the design shown,* the piston is provided with a **loose top** A secured to the **body** C by six square-necked studs. Fracture of such a top is unlikely; should it occur the cost of renewal is small. The central space in the loose top is packed with asbestos, retained by a brass cover-plate B; by this means the underside of the piston is kept relatively cool and oil splashed upon it is not carbonized.

The *body* of the piston is parallel, but the top tapers slightly to allow for expansion. Six C.I. rings of square section are provided. The bottom ring is recessed on the face and acts as a scraper for oil, drain holes (not shown) into the interior being provided at the bottom of its groove. The **gudgeon pin** E is of nickel steel, hardened and ground; it is a driving fit in the piston and is secured by two steel **cotters** D, these in turn being held by set screws. The set screws must be locked in position, and the design of a suitable device is left as an exercise for the student.

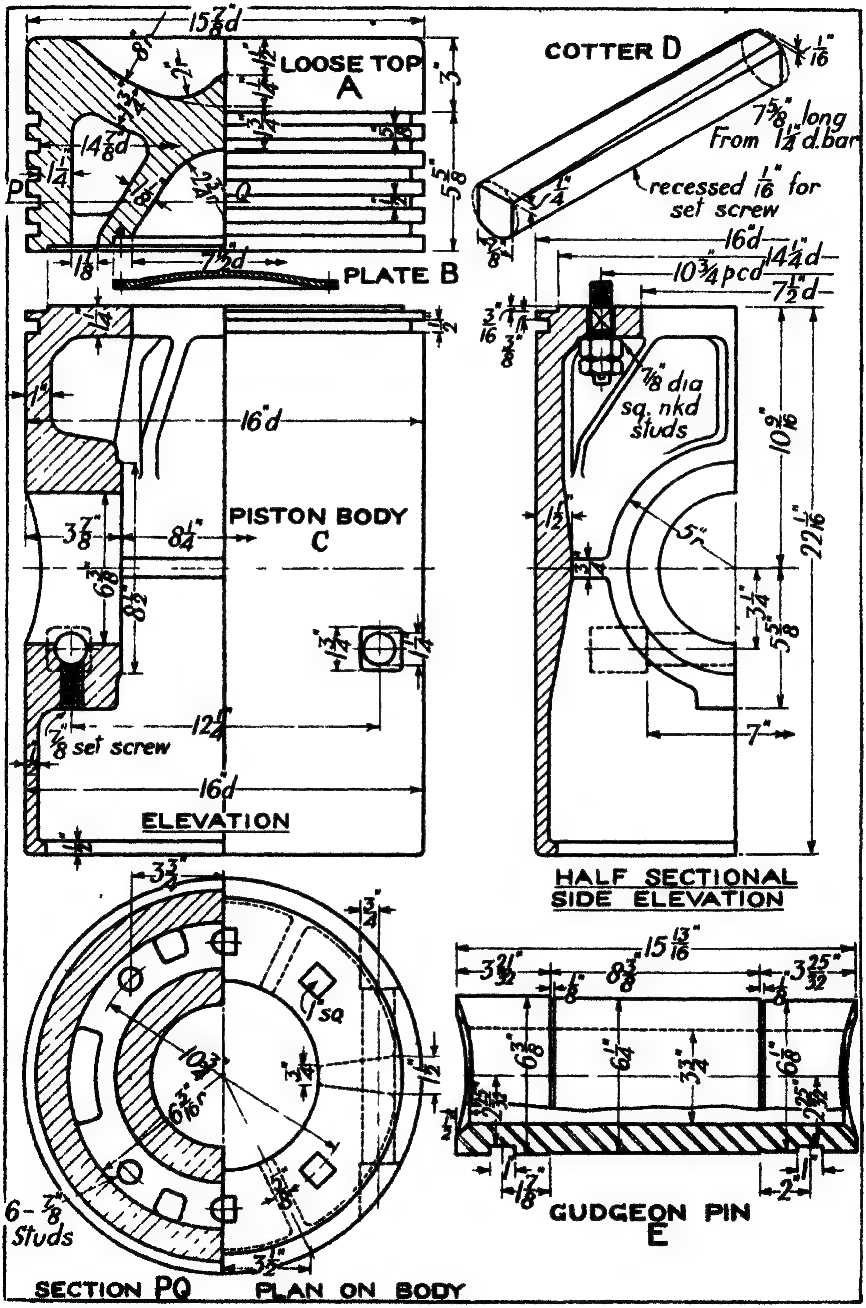
**Proportions.**—These are based almost entirely upon experiment and successful practice. The piston shown is suitable for a stationary four-stroke engine developing 80 h.p. per cylinder at a speed of 240 revs./min. Similar designs are used for cylinders up to 20″ dia.; above this size pistons are usually cooled, by oil or water. The thickness of the crown increases rapidly with the diameter and varies from $1\frac{1}{4}''$ for 10″ dia. to $4\frac{1}{2}''$ for 24″ dia. The following proportions are typical, D being the diameter of the cylinder:—Thickness of piston at bottom of rings 0·075D; length below rings 1·3D to 1·6D; distance above rings = thickness of crown; thickness of skirt 0·03D; ring section 0·03D; gudgeon pin diameter 0·4D, bearing length 0·5D. The axis of the pin is arranged at, or a little above, the centre of the part below the rings.

## EXERCISES

(1) Draw, quarter size, the following undimensioned views of the complete piston, with the component parts assembled: sectional elevation on axis of pin; external elevation on end of pin; sectional plan, one half on the top of the piston body and the other through the pin axis. Use your own judgment for the proportions of details not dimensioned and design a locking device for the cotter set screws.

(2) Prepare, half size, working drawings of a solid piston (i.e. without a loose top) suitable for a cylinder 10″ bore. Adhere to the form and proportions given here but delete all webs. Fit 5 rings.

* The original design has been simplified somewhat to provide the drawings shown: e.g. in the former, the crown is recessed to clear valves, the lower part of the piston is ribbed, openings are provided in the piston skirt for dealing with the set screws, and tapped holes are made for lifting screws.


LOOSE TOP
A
PLATE B
COTTER D
7⅝" long
From 1¼" d. bar
recessed 1/16" for
set screw
7/8" dia
sq. nkd
studs
PISTON BODY
C
7/8 set screw
ELEVATION
HALF SECTIONAL
SIDE ELEVATION
1" sq
6-7/8"
Studs
GUDGEON PIN
E
SECTION PQ
PLAN ON BODY

**Piston Rods** are subjected principally to two sets of forces, (1) the pressure load on the piston, (2) the inertia forces produced by the reciprocating masses: at the dead centres, where the piston load is usually a maximum, these forces are in opposition. The stress in the body of the rod is alternately tensile and compressive, and in design work it is usual to base the diameter of the rod on the max. piston load, using a low value for the working stress—for steel, from 2000 to 4000 lb./in.$^2$.* The screwed end of the rod is in tension only and a stress value of 7000 to 8000 lb./in.$^2$ is permissible. If the diameter of the screw is based on this value and a taper of 1 in 6 allowed where the piston fits the rod, the body of the rod will usually be found sufficiently strong. The diameter of Diesel engine piston rods (where used) is approx. 0·3 of the cylinder bore.

The rod ends are provided either with collars or shoulders. These are left just clear of the piston so that excessive tightening of the nut would bring them into contact and prevent a split piston. When the rod is machined after wear the taper is not encroached upon.

**Stuffing Boxes** are necessary to prevent leakage of the working fluid where the rod passes through the cylinder end. Fig. 1 shows the simplest type of box, in which soft packing† is compressed in an annular space around the rod by a gland drawn inwards by nuts and studs. The proportions vary considerably and the following can be regarded only as a rough guide; $d$ is the diameter of the rod:

$D = 1{\cdot}25d + 0{\cdot}6''$; $L_1 = d + 1''$;
$L_2 = 0{\cdot}5L_1$; $S = 0{\cdot}2d + 0{\cdot}2''$;
$P = 2d + 1{\cdot}1''$; $T = 1{\cdot}25S$;
$W = 1{\cdot}75d + 1''$.

The depth of the packing space $L_1$ should properly be based on the fluid pressure, and Unwin's rule, for steam, is

$$L_1 = \{d(p + 100) \div 170\} + 0{\cdot}8'',$$

where $p$ is the steam pressure in lb./in.$^2$.

Usually a neck ring is fitted to the box and a bush to the gland, both of G.M., as in fig. 2. These can be renewed cheaply after wear. When the piston rod ends are larger than the working part of the rod the bushes must be split longitudinally. The length of the neck ring

$$N = 0{\cdot}5d + 0{\cdot}3'',$$

and the thickness of either bush

$$t = 0{\cdot}05d + 0{\cdot}1''.$$

**Metallic Packing** is now largely used for piston rods. In the design shown in fig. 5 whitemetal half-rings W of wedge section, separated by gunmetal rings R and carried in a cage C, are forced against the rod by the pressure of the gland G, acting through a ring of special soft packing S and a keep ring K. The soft packing is usually impregnated with a lubricating substance. Details of the rings are shown in figs. 3 and 4. Two screwed holes $\frac{3}{16}''$ dia. are provided in each ring for its withdrawal. The whitemetal rings are cut along a diameter; $\frac{1}{8}''$ dia. screwed dowels in one ring enter the lifting holes in the other and keep the divisions of adjacent rings out of line. In many designs springs are employed to give the necessary axial pressure on the metal rings.

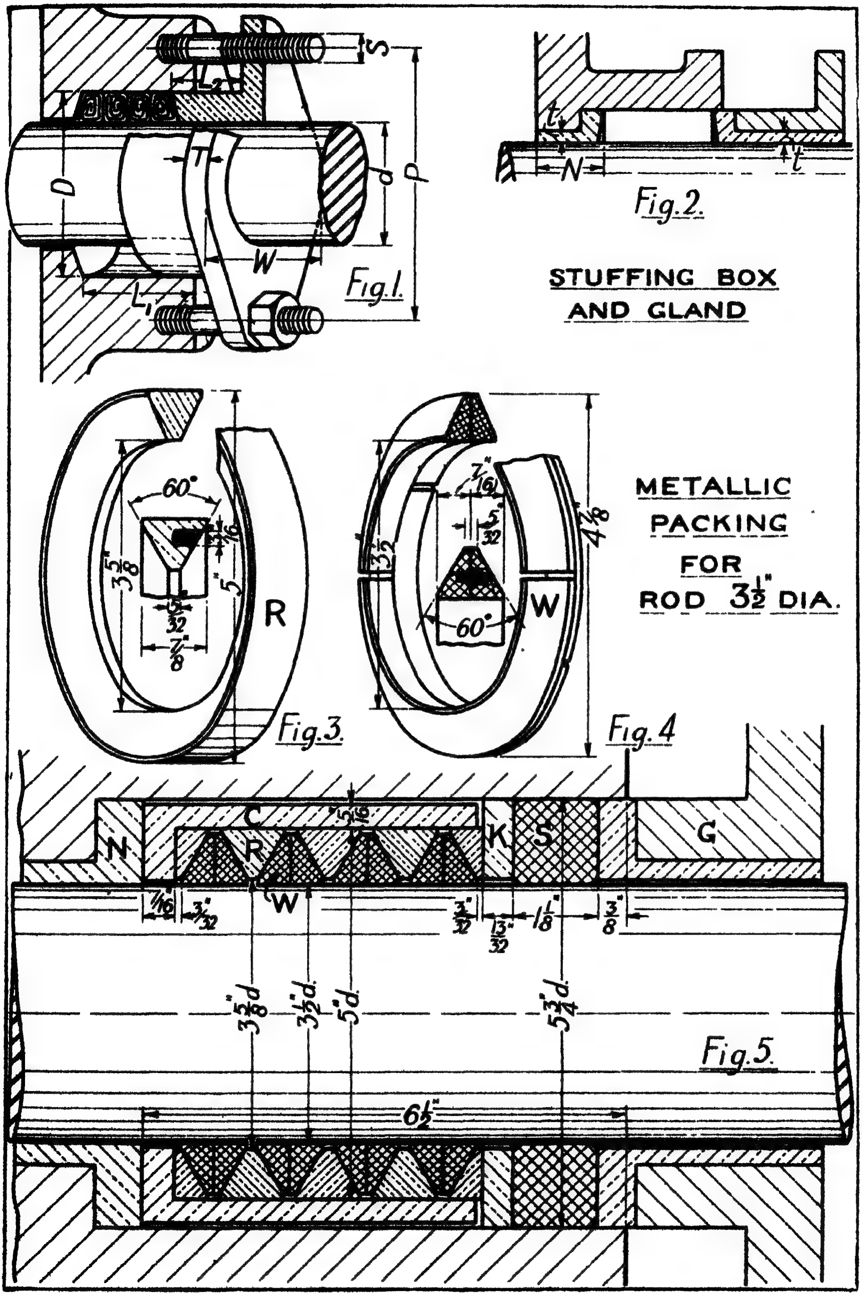
## EXERCISES

(1) A piston rod 2″ dia. passes through the end of a C.I. cylinder $\frac{3}{4}''$ thick, working pressure 100 lb./in.$^2$. Draw a sectional elevation and end view, full size, of a suitable projecting stuffing box and C.I. gland, with G.M. neck ring and bush. Thickness of stuffing-box metal $\frac{3}{4}''$.

(2) The gland G for the metallic packing in fig. 5 is circular, dia. $9\frac{1}{2}''$, and is secured by three 1″ studs. Draw the following views, full size, and dimension them: elevation, half in section on the axis and half on the outside of the rings; end view, half on the gland and half showing the keep plate removed.

* The Rankine-Gordon strut formula was at one time used in designing piston rods, but it would appear more rational to base the proportions on the fatigue stress of the metal.

† Stuffing boxes for steam are commonly packed with asbestos fibre, while gasket (hemp and cotton rope) is used for water.


STUFFING BOX
AND GLAND
METALLIC
PACKING
FOR
ROD 3½" DIA.
Fig.1.
Fig.2.
Fig.3.
Fig.4
Fig.5.
S
L₂
T
D
d
P
W
L₁
t
N
60°
R
W
C
K
S
G
6½"
3⅝"d.
3½"d.
5"d.
5¾"d.

**Connecting Rods** are subjected to both axial and transverse forces: the former are due to the piston load and the longitudinal inertia effect; the latter to the transverse component of the inertia of the rod itself. The effect of journal friction (which is to divert the line of thrust from the centre line of the rod) has been ignored here. The piston load and the acceleration of the moving masses vary during the stroke, and the determination of the position when the straining forces are most detrimental to the rod, also the subsequent calculation of the range of stress, are problems of some complexity.

Rods for high-speed engines are made of I section to give greater resistance to bending in the plane of oscillation and to reduce weight. In engines running at moderate speeds the rods are circular. The ends of the rod, usually called the "big" and "small" ends, have a variety of forms dependent on the type of bearing used at the crank and crosshead. The ratio L : R (fig. 1) varies from 4½ to 6.

**Diesel Engine Connecting Rod.**—The drawings show particulars of a steel rod for a 4-stroke engine developing 80 b.h.p. per cylinder at 240 revs./min., stroke 19″. A suitable piston (16″ dia.) for this engine is shown on p. 137. The rod is circular and parallel, and is 4½ cranks long. The upper bearing is of Phos.B., the lower one of W.M. in C.S. steps. Thin liners are provided under the tee end for compression adjustment, and between the big-end steps to take up wear. Oil is forced through the crank into the big-end bearing; it enters staggered circumferential oil grooves, and then passes through a hole in the rod to the gudgeon-pin bearing.

An alternative design of connecting rod is given on p. 200.

**Proportions.**—The diameter of the rod is first settled by empirical rules and then checked for strength. A common proportion for the diameter (the mean diameter if the rod is tapered) is 0·28D, where D is the cylinder bore.

The greatest axial force on the rod is compressive, and may be assumed to be due to a piston load of 500 lb./in.$^2$ acting during a crank rotation of 30° from the top dead centre. If A is the piston area in in.$^2$, and $\theta$ the obliquity of the rod, the axial force $F^* = 500A \div \cos\theta$ lb. Hence the direct compressive stress $f_c = F \div$ area of rod. To this must be added the stress $f_b$ due to bending: its maximum value occurs when the crank and connecting rod are at right angles (see next page), but for a crank angle of 30°, $f_b$ may be taken at approximately half the calculated maximum value. The resultant stress $f = f_c + f_b$; the value of $f$ should not exceed 9000 lb./in.$^2$.†

Of the tensile load applied to the bolts, that part due to piston friction (which may be considerable) is difficult to estimate. In practice the least diameter of the big-end bolts is about 0·12D: the small-end bolts should be as large as space will permit. Taking a maximum piston load of 500 lb./in.$^2$, the bearing pressure at the small end should not exceed 2000 lb./in.$^2$.

## EXERCISES

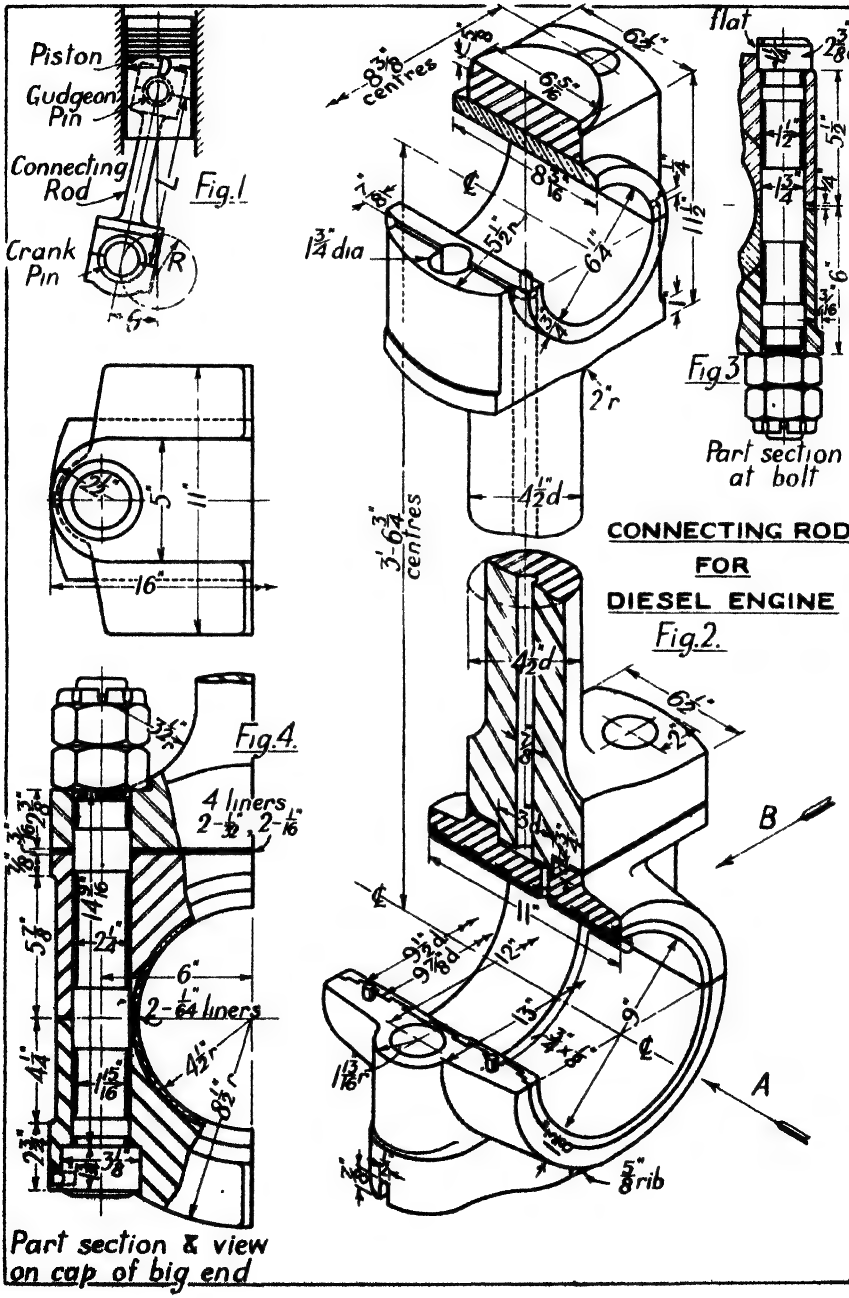
(1) Draw, quarter size, the following views of the connecting rod shown: elevation in direction A with the left-hand portion of each end in section, as in fig. 4; view in direction B; views on both end caps. Arrange the axis of the rod horizontally, breaking the rod to shorten the drawing. Use your own judgment in proportioning undimensioned details. Insert principal dimensions.

(2) Taking a piston pressure of 500 lb./in.$^2$ at dead centre, calculate the direct stress $f_c$ (neglecting inertia effects), and the bearing pressures on each end. What is the safe tensile load for the small end?

*Answers.*—6570; 1970; 1014 *lb./in.*$^2$; 18,600 *lb.*

* Actually F will be reduced by the longitudinal inertia forces.

† It is usual to calculate the range of stress, i.e. the sum of $f$ and the maximum tensile stress, and to limit this to about 12,000 lb./in.$^2$. For a full discussion refer to Purday's *Diesel Engine Design*.


Piston
Gudgeon Pin
Connecting Rod
Crank Pin
Fig. 1
Fig. 3
Part section at bolt
CONNECTING ROD FOR DIESEL ENGINE
Fig. 2.
Fig. 4.
4 liners
Part section & view on cap of big end
centres
B
A

**Locomotive Connecting Rods** are steel forgings of rectangular or I section, usually about six cranks in length. In the design shown the rod is of uniform width and flange thickness, but the depth decreases towards the small end. The bearings are secured to the ends of the rod by steel straps, adjustment for wear being given by cotter and wedge. The strap bolts are slightly tapered to ensure a good fit and to facilitate removal. The small-end steps are of G.M.; those for the large end are of G.M. with whitemetal pads; full details of the large-end bearing are given on p. 192. The cotters bear on the ends of the rod and on the steel wedges: they are clear of the straps. The small end is frequently forged solid, as shown in the small sketch, and fitted either with a G.M. bush or with steps adjustable for wear as shown on p. 195

**Proportions.**—The dimensions of the rod are based upon empirical rules and are checked finally for strength. To the stresses produced by the axial loads must be added those resulting from the transverse inertia forces. It is usual to determine both stresses when the crank and rod are at right angles: the inertia of the reciprocating parts is then sufficiently small to be neglected. Let R = crank radius in feet; L = length of rod (centre to centre) in feet; N = crank speed in revs./min.; $w$ = density of material in lb./in.$^3$.

It may be shown that for a rod of uniform section the bending moment is a maximum at a section (S) of the rod about 0·4L from the crank-pin axis.* The value of this maximum bending moment is:

$$M_{max.} = (0{\cdot}00314\,AN^2L^2Rw)\text{ lb. in.}, \quad (1)$$

where A is the sectional area of the rod in in.$^2$.

If the rod is not uniform, the value of $M_{max.}$ is given, less accurately, by substituting $A_m$, the *mean* sectional area of the rod, for A in (1).

Hence the stress $f_b$ due to bending is approx.:

$$f_b = (M_{max.} \div Z)\text{ lb./in.}^2,\dagger \quad (2)$$

where Z is the modulus in in.$^3$ of the section S.

The axial force (F) on the rod depends on the effective steam pressure ($p$ lb./in.$^2$) in the cylinder, and on the obliquity $\theta$ of the rod. The former must be either estimated or taken from an indicator diagram. For a cylinder D″ dia. the axial force is:

$$F = \tfrac{1}{4}\pi \,.\, D^2 \,.\, p \div \cos\theta\text{ lb.} \quad (3)$$

Hence the stress $f_t$ (or $f_c$), due to the axial load, at section S, is:

$$f_t = F \div A_1\text{ lb./in.}^2, \quad (4)$$

where $A_1$ is the area in in.$^2$ of section S.

The resultant stress $f = f_b + f_t$. The stress is alternately tensile and compressive and the range is therefore $2f$. For locomotive rods $f$ may be 7000 lb./in.$^2$.

The proportions of the ends are largely governed by the size of the bearings. In settling the latter, pressures of 3000–4000 lb./in.$^2$ for the small end and 1000–1500 for the large end are common. The strap is in tension and is usually weakest at the bolts; the bolts are in double shear; and the cotter is in compression: the stresses in these should not be excessive (see Exercise 2).

**Details of a connecting rod suitable for a slow-speed stationary engine are given on pp. 194 and 195.**

## EXERCISES

(1) Draw, half size, the following dimensioned views of the small end and bearing: elevation, upper half in section; plan; end view from right to left.

(2) If the axial load on the rod is 29,270 lb., calculate the stresses in the following parts: (*a*) smallest section of rod, (*b*) bolts, (*c*) straps (at bolt holes), (*d*) cotter; also calculate the bearing pressure at the small end.

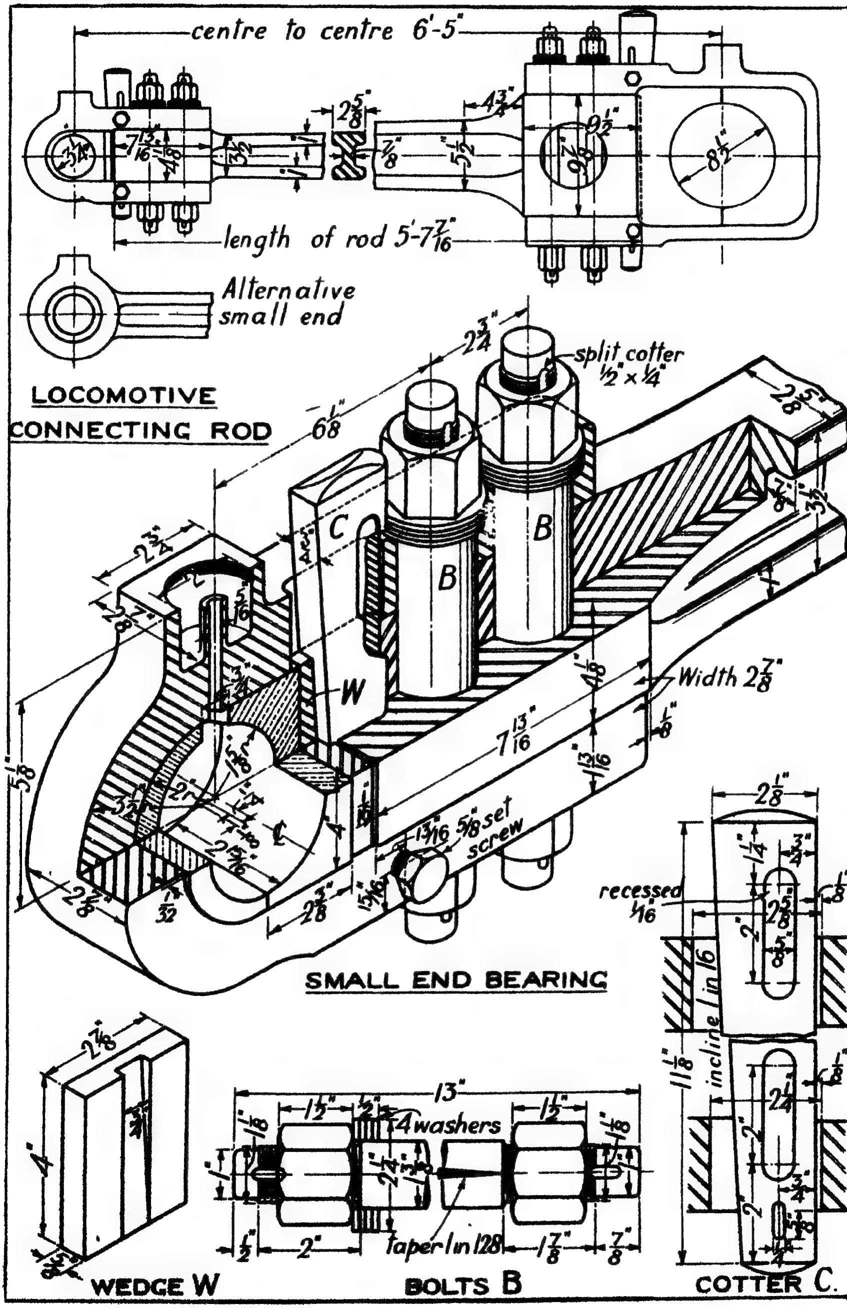
*Answers.*—(*a*) 4940, (*b*) 4940, (*c*) 5400, (*d*) 9760; 3070 *lb./in.*$^2$.

(3) The rod shown is suitable for a cylinder 17½″ dia. × 26″ stroke, speed 300 revs./min. If the effective steam pressure when rod and crank are at right angles is 120 lb./in.$^2$, calculate the maximum stress in the rod. Assume that the section (S) at which the bending moment is greatest is 5″ deep.

*Answer.*—$f_b$ = 2850, $f_t$ = 3730, $f$ = 6580 *lb./in.*$^2$.

* It is assumed that the influence of the ends of the rod is small. Refer to *Machine Design* by Unwin and Mellanby for proof.

† For a solid circular steel rod dia. $D_1$ this gives $f_b = 0{\cdot}007\ N^2L^2R \div D_1$.


centre to centre 6'-5"
length of rod 5'-7 7/16"
Alternative small end
LOCOMOTIVE CONNECTING ROD
split cotter 1/2" × 1/4"
Width 2 7/8"
5/8" set screw
SMALL END BEARING
recessed 1/16
incline 1 in 16
4 washers
taper 1 in 128
WEDGE W
BOLTS B
COTTER C.

**The crosshead** of an engine is a sliding piece which connects the ends of the piston and connecting rods. A great variety of designs have been evolved and two of the principal types are included herein: the locomotive and the marine. In all designs the crosshead must provide (*a*) a rigid connexion for the piston-rod end, (*b*) a journal for the small-end bearing of the connecting rod, and (*c*) bearing surfaces to suit the guides between which it reciprocates. The bearing surfaces are usually plane and may be provided on the crosshead itself or on slide blocks attached to it. They are made of large area to reduce the bearing pressure, which varies in practice from 30 to 100 lb./in.$^2$.

By applying the triangle of forces to the crosshead (the three forces being the piston load P, the thrust F in the connecting rod, and the reaction R at the guide) the force on the guide is given by $P \tan\theta$, where $\theta$ is the obliquity of the connecting rod. For double-acting engines this force acts chiefly in one direction for the same way of rotation of the crank. Hence engines which are to revolve mostly in one direction may have small guide areas for the reverse motion.

**Crosshead for Locomotive.**—An elevation and end view of a crosshead and its guides are shown in fig. 1. The crosshead without its fittings is shown, partly sectioned, to a larger scale in fig. 4. Figs. 2 and 3 show respectively small-scale details of one of the two slide blocks, or shoes, and the crosshead pin.

The shoes fit over dovetailed circular projections on the crosshead. They are first placed in position transversely and turned into alignment about the central boss. The piston-rod end is coned and cottered in the socket; the axial position of the cotter hole is indicated in fig. 4, but actually the cotter is inclined at 30° to the horizontal, as in fig. 1. The crosshead pin is tapered where it passes through the cheeks of the crosshead, and keyed at the large end.

The crosshead is a steel casting and the shoes are of C.I. with faces lined with W.M. The pin and all other parts are of steel. Lubricating and other fittings have been omitted to simplify the drawings.

**Proportions.** — The form of the crosshead is complicated, and the only parts which can be proportioned by calculation are the guide blocks, piston-rod socket, and crosshead pin. In determining the guide-block areas it is usually assumed that the maximum load on the guides $= P \div r$, where P is the maximum piston load and $r$ the ratio length of connecting rod/radius of crank. The socket joint may be dealt with as on p. 66. The crosshead pin is in double shear and will be found usually to be amply strong if proportioned to give a reasonable bearing area.

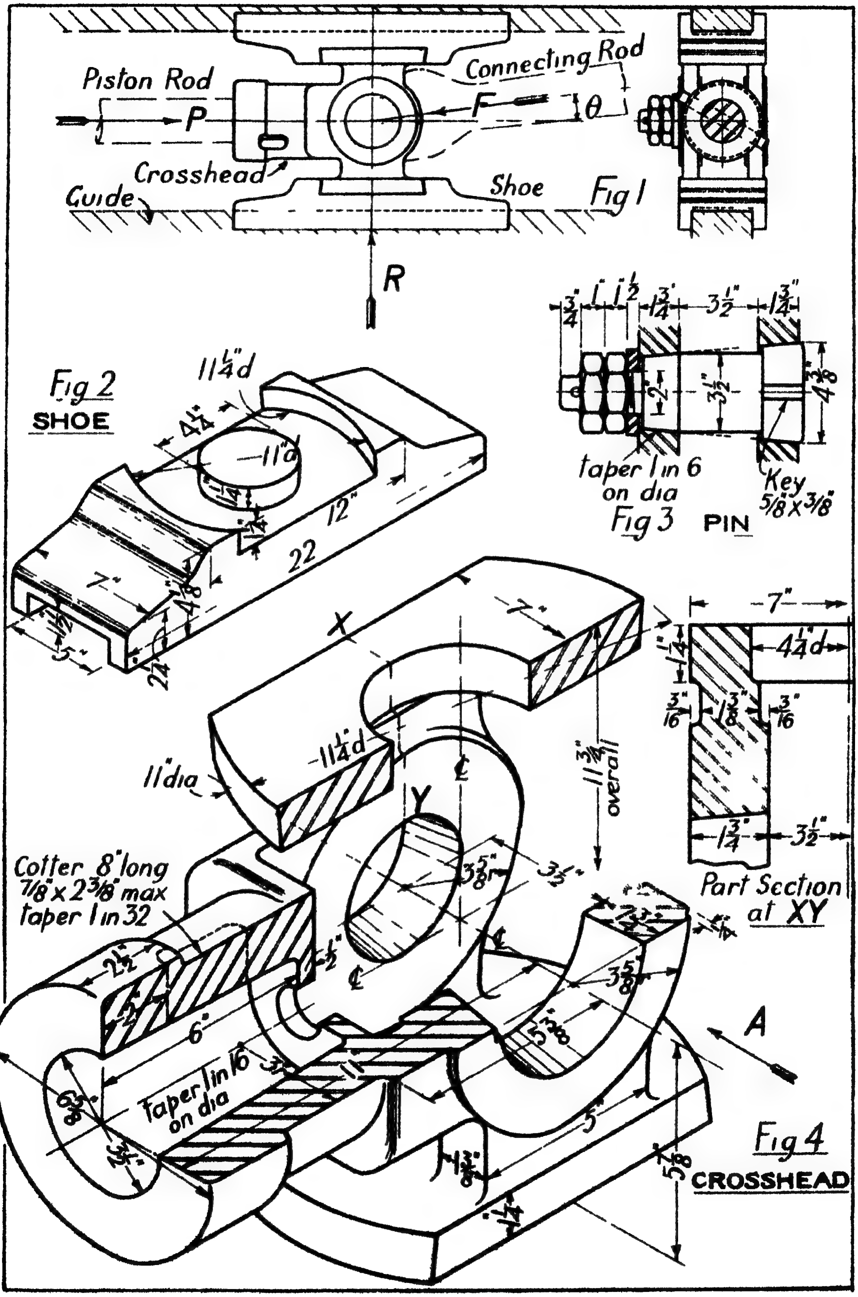
**Marine-type Crosshead.**—This is fully illustrated on p. 193.

## EXERCISES

(1) *Draw, half size, the following dimensioned views of the crosshead complete with shoes and pin, but omitting the cotter hole; show five whitemetal pads, each $3'' \times 4\frac{1}{2}'' \times \frac{1}{2}''$ thick, dovetailed into each shoe face and flush with the C.I. surface: elevation in direction A, with the upper half in section; plan, with one half in section on the piston-rod axis; end view from right to left. Use your own judgment where dimensions are omitted.*

(2) If the piston rod sustain a thrust of 30,000 lb., calculate: (*a*) the guide pressure (taking $r = 6$), (*b*) the stress at the weakest part of the socket (i.e. at the cotter hole), (*c*) the shear stress in the crosshead pin.

*Answers.*—(*a*) 45·5, (*b*) 1700, (*c*) 1560 *lb./in.*$^2$ (*approx.*).


Piston Rod
Connecting Rod
P
F
θ
Crosshead
Guide
Shoe
Fig 1
R
Fig 2
SHOE
11¼"d
4¼"
11"d
12"
22
7"
5"
Fig 3 PIN
taper 1 in 6 on dia
Key 5/8"x3/8"
Part Section at XY
X
Y
11"dia
11¾" overall
Cotter 8" long 7/8"x 2 3/8" max taper 1 in 32
6"
Taper 1 in 16 on dia
A
Fig 4
CROSSHEAD

**Steam-engine Cylinders.**—These are invariably castings, either in iron or steel, of various forms, the essentials in each being the barrel, in which the piston slides, the valve chest, and the supporting brackets. Passages are formed in the casting to convey steam from the valve chest to the cylinder and from the cylinder to the exhaust. The casting is somewhat complicated, even for small cylinders, and its design is governed by such factors as the type of valve used (flat, piston, or poppet), the relative positions of steam inlet and exhaust outlet, and positions of supports. Large cylinders are usually steam jacketed and fitted with liners.

**Single C.I. Cylinders.**—For a given horse-power, the dia. D and the length of stroke L are settled by the formula, horse-power $= p_m.L.A.N/33{,}000$, where $p_m$ = mean effective pressure during the stroke (which must be estimated*), L = stroke in feet, A = cylinder area in in.$^2 = \frac{1}{4}\pi . D^2$, N = number of working strokes/min. A value for D or L must be chosen arbitrarily, there being no standard relationship between them.

The barrel must be made thick enough to give a sound and rigid casting and to admit of reboring after wear: if these requirements are met it will usually be over strong to resist the steam pressure. An empirical rule for the thickness of C.I. cylinders without liners, due to Unwin, is

$$t = 0{\cdot}00015D(p + 55) + 0{\cdot}5'',$$

where $t$ = thickness in inches and $p$ = steam pressure in lb./in.$^2$. The thickness of the cylinder flanges $= 1{\cdot}3t$, and the thickness of the metal in the valve chest and passages $= 0{\cdot}7t$.

The mean velocity $v$ of the steam through the passages is given approximately by $v = VA/a$, where V = mean piston speed in ft./sec., A = cylinder area, and $a$ = port area, both in in.$^2$. The area $a$ should be such that the velocity $v$ does not exceed 110 ft./sec.

**Cylinder for Locomotive.**—The drawings show details of a C.I. locomotive cylinder, bore $18\frac{1}{2}''$, stroke $26''$, working steam pressure 160 lb./in.$^2$, fitted with a piston valve. Fig. 1 is a longitudinal section and figs. 2 and 3 are transverse sections to a smaller scale. It will be evident that additional sections are required to describe the cylinder completely. The original design has been simplified somewhat by the omission of by-pass valves and passages, webs, &c.

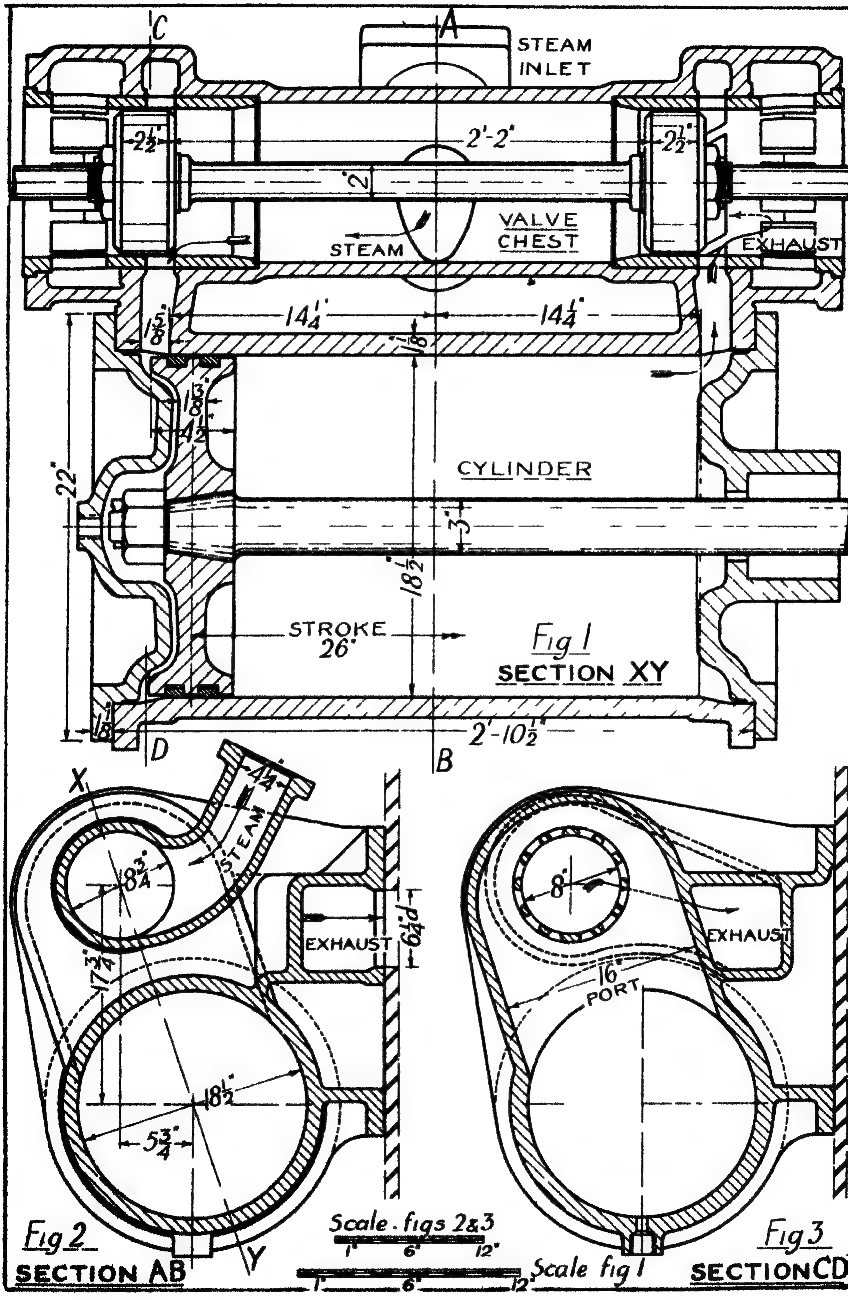
The valve chest is fitted with liners pressed into place and held by the end covers (not shown): details of the piston valve and liners are given on p. 187. Steam is admitted between the piston heads, and exhaust passages lead from the ends and join in a common outlet. The cylinder covers are of dished form and follow the contour of the piston. They are usually webbed for strength. Brackets are cast on the cylinder for side attachment to the engine frame.

**C.I. Cylinder for Vertical High-speed Engine.**—This is shown on pp. 210 and 211. It should be referred to and compared with the design opposite.

## EXERCISE

Draw, quarter size, a longitudinal section of the cylinder with its piston and valve, as shown in fig. 1, adopting the dimensions given but using your own judgment in proportioning and arranging undimensioned parts generally. Use the following data. Piston: least thickness $1\frac{3}{8}''$, rings $1'' \times \frac{1}{2}''$. Covers: thickness, outer $\frac{3}{4}''$, inner $1''$; secured to cylinder by $\frac{7}{8}''$ studs; clearance at ends $\frac{3}{8}''$. Valve chest: metal $\frac{3}{4}''$ thick, steam port $1\frac{3}{8}''$ wide. Details of the valve are given on p. 179. Find by calculation a suitable number of cylinder cover studs.

* Theoretically $p_m = p\left\{\frac{1 + \log_e r}{r}\right\} - p_b$, where $p$ = admission pressure, $p_b$ = back pressure, $r$ = ratio of expansion.


A
C
STEAM
INLET
2½"
2'-2"
2½"
2"
VALVE
CHEST
STEAM
EXHAUST
1⅝"
14¼"
14¼"
1⅛"
1⅜"
4½"
22"
CYLINDER
3"
18½"
STROKE
26"
Fig 1
SECTION XY
1⅛"
2'-10½"
D
B
X
4¼"
STEAM
8¾"
EXHAUST
6¼"d
17¾"
18½"
5¾"
Y
8"
EXHAUST
16"
PORT
Fig 2
SECTION AB
Scale. figs 2&3
1" 6" 12"
Scale fig 1
1" 6" 12"
Fig 3
SECTION CD

**De Laval Turbine.**—In this turbine jets of steam from nozzles arranged in the casing are directed against a ring of crescent-shaped blades fixed to the rim of a single wheel designed to revolve at high speed. The steam expands freely in the nozzles, where its heat energy is converted into kinetic energy, and is delivered at low pressure but high velocity against one side of the ring of blades, as in fig. 1, the steam passing to exhaust at the other. In its passage across the concave faces of the blades the velocity of the steam, relative to the blades, remains almost constant in magnitude, and the action on the wheel is one of pure impulse.

**Construction.**—The wheel shown is designed to develop 30 h.p. at a speed of 20,000 revs./min. This speed of rotation gives rise to high stresses in the wheel, both radial and tangential, which in practice are allowed to reach a maximum total of about 11 tons/in.$^2$. The thickness $t$ of the wheel at radius $r$ is given by the equation $t = C \div r^n$, where C is a constant and $n = 2$ for De Laval wheels. Any surplus material at the rim increases the disruptive forces without adding to the strength of the wheel: hence the section is kept as light as possible towards the rim. A reduction in thickness is made just under the rim so that in the event of the speed becoming excessive the portion beyond the reduced section would fly off in pieces; the unbalanced wheel would then foul the casing and be brought to rest. The working stress in the "safety section" is about 16 tons/in.$^2$.

The wheel rim is slotted to receive the enlarged ends of the blades, which are a press fit in the slots. The bulbs are very slightly riveted over in fixing, but not so much as to prevent the blades from being driven out for replacement. There is very little *axial* thrust on the blades, the inlet and outlet angles being equal. The rectangular portions at the blade tips form an elastic ring.

The bush, fig. 5, fits an enlarged tapered portion on the spindle, fig. 3, and is secured to it by a central rivet. The wheel hub is bored to fit the bush, also slightly tapered, and is secured by the circular nut shown, the end collar of the bush fitting the recessed portion at the end of the hub.

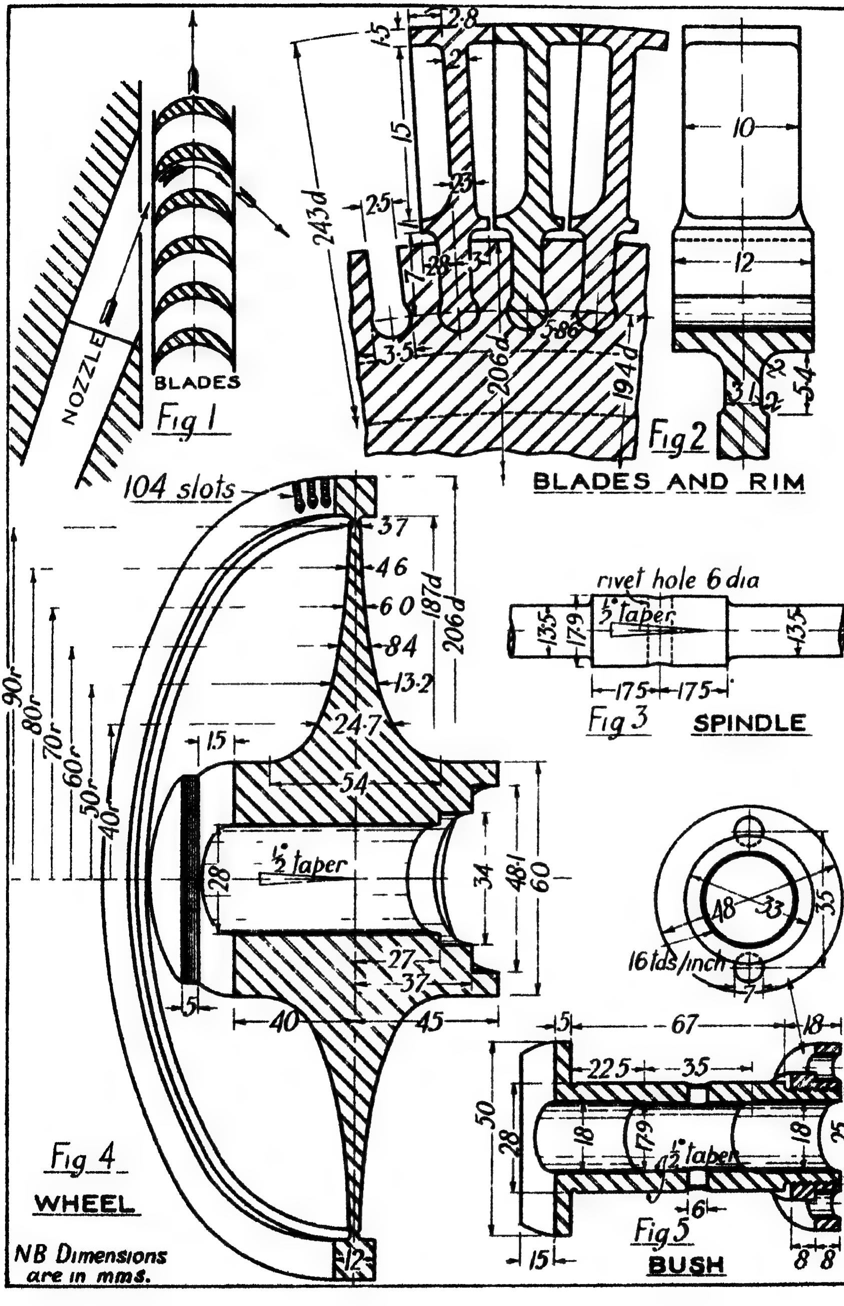
The spindle is thin enough to be flexible, and is run much above its critical speed. On starting, the wheel runs unsteadily; it runs true after the critical speed has been passed. So designed, the wheel may be run at high speeds without vibration. The torque is transmitted through single-reduction double helical gearing, the driven shaft running at 2000 revs./min.

**Materials.**—These are all of the highest possible grade. The wheel is of nickel chrome steel heat treated, and the blades are of stainless steel. The spindle is of 0·6 per cent C steel, and all other parts are of mild steel.

**Units.**—De Laval turbines are made in standard sizes, the smallest giving 5 b.h.p. at 30,000 revs./min., and the largest 600 b.h.p. at 9500 revs./min.: the respective mean blade speeds are 515 and 1550 ft./sec. When using dry saturated steam at 200 lb./in.$^2$ and exhausting to a 27" vacuum, the steam consumption of the smallest unit is 33·9 lb. per b.h.p. per hr., and of the largest 11·8 lb. per b.h.p. per hr.

## EXERCISE

Draw, full size, the following views of the blank wheel in position on its bush and spindle: elevation, half in section on the axis; end view, one half from each side. Also draw, 5 times full size, views corresponding to those in fig. 2 showing three blades in position. Fully dimension the drawings. Note: dimensions are in millimetres.


NOZZLE
BLADES
Fig 1
243d
206d
194d
Fig 2
BLADES AND RIM
104 slots
187d
206d
90r
80r
70r
60r
50r
40r
1/2° taper
Fig 4
WHEEL
NB Dimensions are in mms.
rivet hole 6 dia
1/2° taper
Fig 3
SPINDLE
16 tds/inch
1/2° taper
Fig 5
BUSH

**Combined Impulse Turbines.**—When the energy of a steam jet is used in one impulse stage only, as in the De Laval turbine, it is not easy to convert all the available energy of the steam into useful work; further, for maximum efficiency in a single stage, an exceptionally high blade speed is required. For these reasons turbines are "compounded", and the energy of the steam is extracted in stages at a slower blade speed.

**Impulse Type Turbine.**—*B.H.P.* 2400; *speed* 2650 *revs./min.*; *initial steam pressure* 200 *lb./in.*$^2$.

The expansion of the steam is carried out in eight stages, the impulse wheels being keyed to a common shaft. Each wheel revolves in a separate chamber formed by fixed diaphragms which partition off the turbine casing, the shaft passing through glands in the diaphragms. Inclined nozzles are arranged in the diaphragms; in its passage through them the steam falls in pressure and gains in velocity, and is redirected on to the blades of the next wheel. There is no appreciable change of pressure as the steam passes through the blades of any one wheel: hence the pressure is uniform throughout each chamber, and there is no axial steam thrust on the wheel discs. The first-stage wheel has *two rows* of blades; a row of fixed blades is arranged between them to redirect the steam on to the second moving row. This stage is typical of the Curtis turbine, being compounded for velocity.* The nozzles for the first-stage wheel are arranged in the turbine casing and are controlled by valves.

**Impulse Wheels and Shaft.**—Eight forged-steel wheels are pressed on the shaft, four from each end, over feather keys. To facilitate assembly the shaft is stepped and successive feathers are reduced in width. The shaft steps and wheel bores are tapered 1 in 48 on the dia. for fitting, and circular nuts screwed on each end of the shaft lock the wheels in position. Each wheel is serrated on one hub face to make a steam-tight joint against the plane face of the next wheel. Grooves as in fig. 2 are cut in the hubs of all wheels except the first for the insertion of gland strips (see later). Three $\frac{7}{8}''$ dia. tapped holes (not shown) are provided in each disc near the boss for withdrawing the wheel from the shaft and for equalizing the steam pressure on each side. All wheel rims except the first are identical.

The blades are of stainless steel, machined from the solid bar and polished. The roots are enlarged to provide the correct blade spacing and dovetailed to fit the groove in the rim. They are entered through openings in the rim which are subsequently plugged. The tips of the blades pass through a shrouding ring and are riveted over. The principal details of the blades for the first and second stages are given in figs. 2 and 3, there being 250, 286, and 290 blades respectively in the first, second, and third moving rows.

*All wheels have the same mean blade dia.*, 36″.

## EXERCISE

Draw, half size, the following views of the first- and second-stage wheels in position on the shaft, showing the shaft broken at the left of the second wheel and at the right of the nut: sectional elevation through the feathers and shaft axis, showing that part of the discs below the hubs broken off; end elevation on the nut, showing about one-quarter of the wheel. Omit the blades in the end view.

The nut is to have 4–$\frac{3}{8}''$ radial holes for a ring spanner and is to be provided with a locking plate, fitting a shallow groove in the wheel and secured to the nut by a square-necked stud: the design of this is left as an exercise for the student. Size of feathers: first stage, $4\frac{1}{8}'' \times 1'' \times \frac{3}{8}''$; second stage, $3\frac{13}{16}'' \times 1\frac{1}{16}'' \times \frac{3}{8}''$. Use your own judgment where dimensions are omitted.

* The provision of a velocity wheel is a very desirable feature, since the steam temperature and pressure may be reduced considerably by expansion in the primary nozzles before being admitted to the cylinder proper.

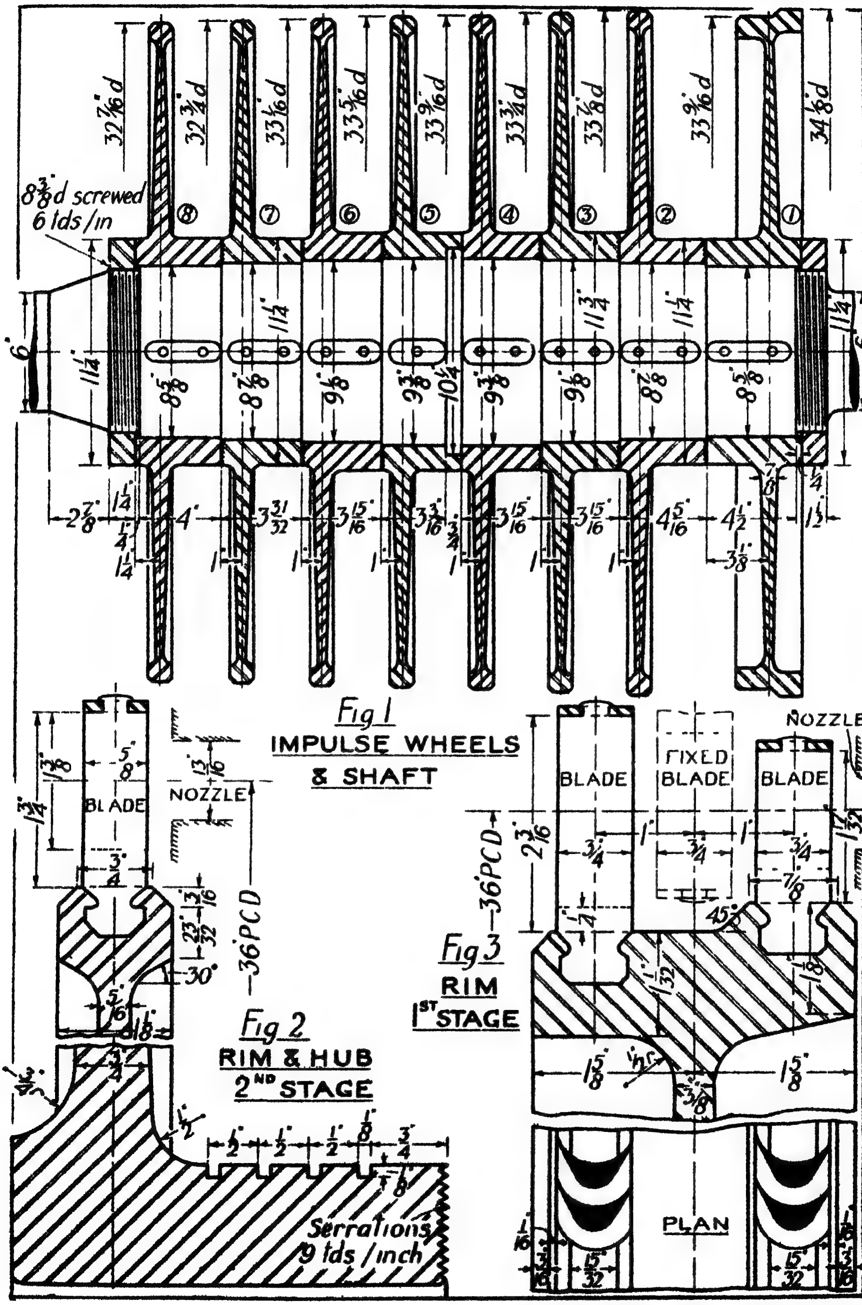



Fig 1 IMPULSE WHEELS & SHAFT

Fig 2 RIM & HUB 2ND STAGE

Fig 3 RIM 1ST STAGE

**Diaphragms for Combined Impulse Turbine** (*see also page* 154).—As stated on page 150, the turbine casing is partitioned off into eight compartments by C.I. diaphragms, seven of which are required. Details of the second-stage diaphragm (i.e. between the 1st and 2nd wheels) are shown opposite and on page 155.

Each diaphragm is split along a diameter. The joint between the halves is spigoted and grooved, as shown in fig. 3. The small sections in fig. 2 show that each half has a spigot on one side and a recess on the other. A semicircular groove $\frac{5}{16}''$ dia. is cut centrally along both spigots and recesses for the insertion of packing. When the halves are in position they butt at the joint and the packing prevents steam leakage from one side to the other. Pins are arranged along the groove to prevent movement of the packing, but these have been omitted in the drawing.

The diaphragms are subjected to an axial load due to the difference in steam pressures on opposite sides—i.e. the difference between the steam pressure in adjacent compartments. They are made slightly conical in form for rigidity, as shown in the sectional view, fig. 1; it should be noted that in this view lines beyond the section itself have been omitted to avoid confusion.

The outer periphery enters a groove a little wider than itself in the turbine casing, and bears against the side of the groove in the direction of the steam flow. The clearance between the rim and the groove is taken up by six brass fitting strips secured to the diaphragm by two $\frac{3}{8}''$ dia. countersunk head brass screws. All diaphragms have the same external diameter, $3'$–$5\frac{7}{8}''$.

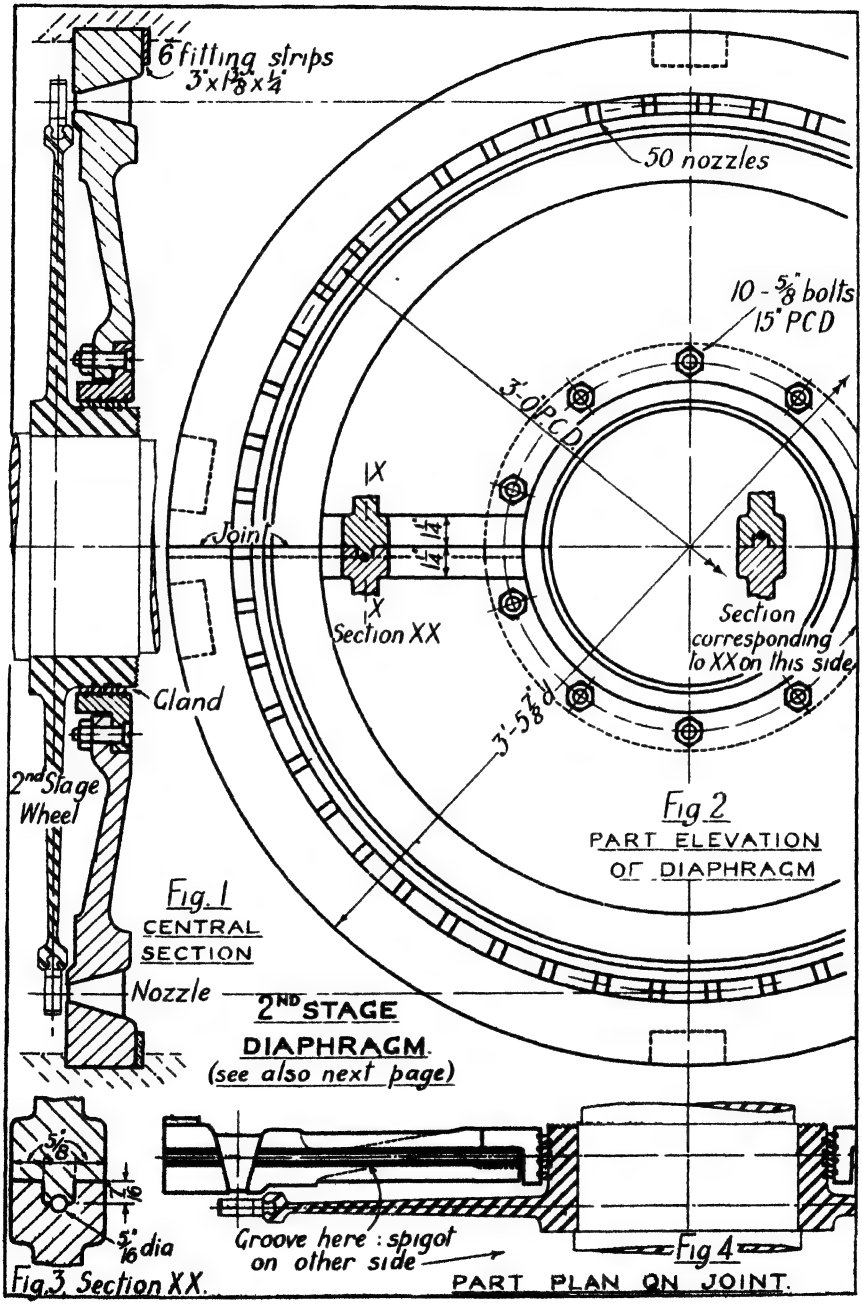
**Nozzles.**—Each diaphragm contains 50 nozzles on a pitch circle $3'$–$0''$ dia. The nozzle plates are of stainless steel, $0{\cdot}116''$ thick, bent to the shape shown on page 155. They are cast in position in the C.I. diaphragm so that the upper and lower edges of the plates are firmly embedded in the surrounding metal. This is shown more clearly on page 155.

Special fitting strips of steel are required where the joint of the diaphragm cuts the nozzle plates, but these, together with the temporary lugs cast on the halves of the diaphragm to facilitate machining, have been omitted from the drawings shown.

**Glands.**—The inner periphery of the diaphragm is provided with glands of the labyrinth type which almost touch the shaft. The flanged packing ring is of C.I. and is split on a diameter: the halves are secured to the diaphragm with steel bolts fitted with naval brass nuts. Grooves $\frac{1}{8}''$ square in section are cut in the shaft and in the bore of the ring—which is $\frac{1}{4}''$ clear of the shaft. In these grooves are driven brass * strips which are afterwards turned to a sharp edge—as shown to a larger scale on page 155. The tips of the brass rings are $0{\cdot}01''$ thick and are given a clearance of $0{\cdot}025''$. If contact takes place between the rings and the shaft no damage will result, as the ring tips wear away easily to a clearance. The labyrinth arrangement effectively prevents undue steam leakage between the shaft and the diaphragms, and avoids the friction losses of the rubbing or contact type of gland.

**Exercises.**—See page 154.

* Zinc, 30 per cent; copper, 70 per cent.


6 fitting strips
3" x 1 3/8 x 1/4"
50 nozzles
10 - 5/8" bolts
15" PCD
3'-0" P.C.D.
X
Joint
1 1/4"
1 1/4"
Section XX
Section corresponding to XX on this side
Gland
3'-5 7/8" d
2nd Stage Wheel
Fig 2
PART ELEVATION OF DIAPHRAGM
Fig. 1
CENTRAL SECTION
Nozzle
2ND STAGE DIAPHRAGM
(see also next page)
5/8"
7/16
5/16" dia
Fig. 3. Section XX.
Groove here: spigot on other side
Fig 4
PART PLAN ON JOINT.

**Details of Diaphragms and Blades.**—The drawings given opposite refer to the second-stage diaphragm of the Impulse-type Turbine, and amplify those shown on the previous page. In the half-section given in fig. 2, lines beyond the actual section have been omitted for clearness. Fig. 4 shows the labyrinth packing to a larger scale.

**Nozzle Plates.**—A nozzle plate (fig. 1) for the diaphragm is cut from a flat sheet of stainless steel 0·116″ thick and bent over a cylindrical block $1\frac{1}{4}$″ rad. to the form shown in fig. 5. The series of 50 plates is then cast in the C.I. diaphragm by a special process, the dotted lines in fig. 2 showing the embedded plate. The exposed area of plate forming the nozzle is defined by chain lines in fig. 1. A developed or expanded section at the pitch line of the second-stage nozzle plates and impulse blades is given in fig. 5.

The nozzle plates of all diaphragms appear as in fig. 5 (on the pitch line), but, as stated earlier, the radial width of the openings is increased for successive stages and longer blades are fitted to the wheels; e.g. the nozzle of the last stage is $2\frac{1}{16}$″ wide at the throat and 3″ wide at the inlet. The method of locating the centre of the arc and of setting out the nozzle plates should be clear from fig. 5.

**Blades.**—The entrance and exit angles of the arc at the front of the blades are determined from a velocity diagram for the steam and blade. The wheels revolve at 2650 revs./min., giving a blade speed at the pitch line U = 416 ft./sec. The steam is directed on to the blades at an angle $\alpha = 12°$, and its velocity V will be taken here as 850 ft./sec.* The velocity of the steam *relative to the blade*, R, is obtained by setting out the velocity diagram as in fig. 3. The theoretical blade angle at entrance is given by $\theta$.†

The speed of the steam as it passes across the blade surface is reduced by about 28 per cent by friction, but it will be assumed here that there is no loss and that the exit velocity $r$ of the steam relative to the blade is equal to R. The exit angle $\phi$ of the blades should be such that the steam leaves in an axial direction to enter the nozzles of the next diaphragm without shock. In the figure $\phi$ has been made equal to $\theta$, and the smaller triangle gives the velocity diagram at the exit side: U is the blade velocity, $r$ the velocity of the steam relative to the blades (taken = R), and $v$ the final absolute velocity of the steam. The direction of the steam at exit is given by $\beta$.

Actually the blades are not quite symmetrical and lie at a slight angle axially as in fig. 3, p. 151. The radius of the arc of the working face is given by $b \div (\cos\theta + \cos\phi)$, where $b$ = width of blade, here $\frac{5}{8}$″.

## EXERCISES

(1) Draw, quarter size, the following views of the second-stage diaphragm *arranged as on the previous page* but showing the gland removed: elevation, as in fig. 2, p. 153; end elevation showing the lower half in section; complete plan on the joint. Fully dimension the drawings.

(2) Draw, full size, an elevation, sectional end view, and plan of the *lower half only* of the gland, showing one bolt in position. Dimension the views completely.

(3) Using the data given, draw the velocity diagram and determine $\theta$, as in fig. 3. Taking $\phi = \theta - 3°$ and $r$ = R, draw the smaller diagram and measure $\beta$. Then set out, full size, an expanded view of the nozzles and blades as in fig. 5, but showing the blades at their correct angle. Thickness of blades 0·21″.,

*Answer.*—$\theta = 23°$, $\beta = 87°$ *approx.*

(4) Draw, full size, a section *at the nozzle* of the diaphragm, as in fig. 2, using the details given in figs. 1 and 5 and assuming in your projection that the radius of the wheel is infinitely large.

* Maximum efficiency is given when U = ½V cos α.

† It was at one time considered important to make the blade entrance angle = $\theta$, but better results are given by increasing $\theta$ by 3° to 5°.

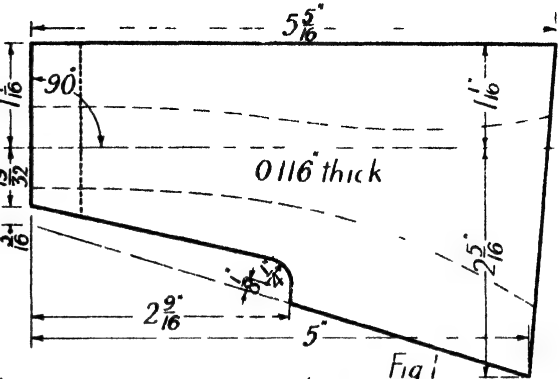



Fig 1
EXPANDED VIEW
NOZZLE PLATE

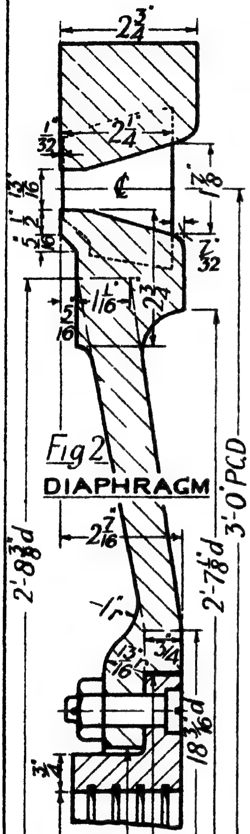



Fig 2
DIAPHRAGM

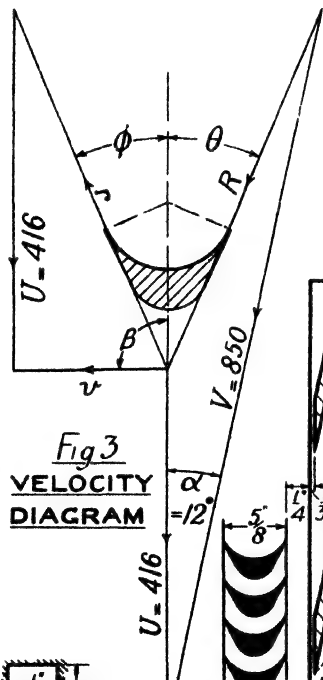



Fig 3
VELOCITY
DIAGRAM

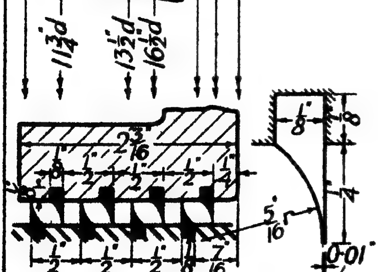



Fig. 4 LABYRINTH
PACKING

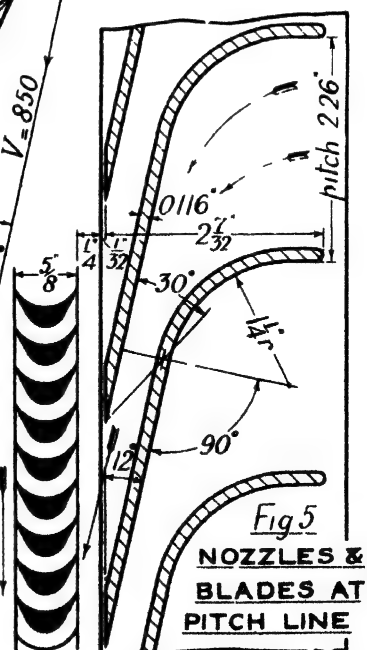



Fig 5
NOZZLES &
BLADES AT
PITCH LINE

The designer can rarely consider only the scientific aspect of design: the economics of production are often all-important. Hence the student should continually attempt the interesting task of changing a design to effect a reduction in cost.

Economical specialist production in quantity is desirable for export trade in such a competitive industry as engineering; for a piece of machinery can be manufactured in many countries in a general way for about the same cost. If, however, it is produced in quantity in a country well adapted for its manufacture, then its production elsewhere, especially if the demand for it there is small, is only possible behind a protective tariff—which raises its cost to users in that country.

The factors making for cheap production and satisfactory service may be grouped as under:—

(1) The article must be economically proportioned.

(2) Materials and methods of fabrication must be the cheapest acceptable.

(3) Hand processes should be reduced to a minimum.

(4) The design should be planned for production.

(5) Standardized parts should be incorporated.

(6) Interchangeability of parts should be arranged.

Progress in design is the result of three insistent demands: from commerce, a reduction in price; from service, a reduction in operating troubles; from science, a reduction in energy losses.

## 1. Economical Proportions.

These are achieved by allowing high stress values—which can only be adopted if loads are known exactly and a precise stress analysis is possible. In the simple designs analysed in this book (e.g. the cottered joint on p. 67), it has been assumed that the stress is uniform over the section. This is often not the case; and the lack of uniformity is particularly important when parts are subjected to cyclic stress variations.

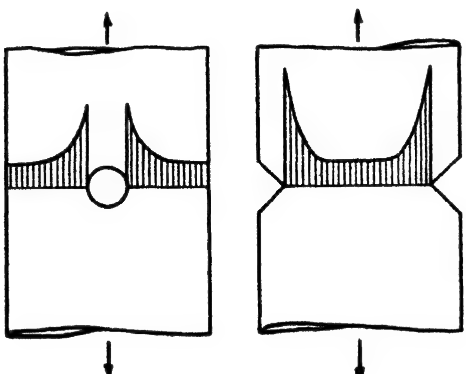


Fig. 1.

*Stress Concentration.*—The effects of a notch or a hole in a circular shaft subjected to simple tension are indicated by the stress diagrams in fig. 1. The important point is that the maximum stress value is higher than the mean; and unless this maximum falls below the elastic limit, local yielding will result. Similar stress concentration is produced at sharp fillets. Hence in any design where high cyclic stress variations are imposed, e.g. at the crosshead of an engine, abrupt changes of section, sharp corners, collars, key-

ways and holes should, wherever possible, be avoided.* (The student should realize that where an analysis along these lines of a successful design would indicate failure, the explanation is probably that local yielding has permitted a complete redistribution of stress.) Fillets of large radius are very desirable in crankshafts, but often cannot be adopted in the usual design without unduly lengthening the shaft. By using *internal* fillets, however, a very satisfactory arrangement results.

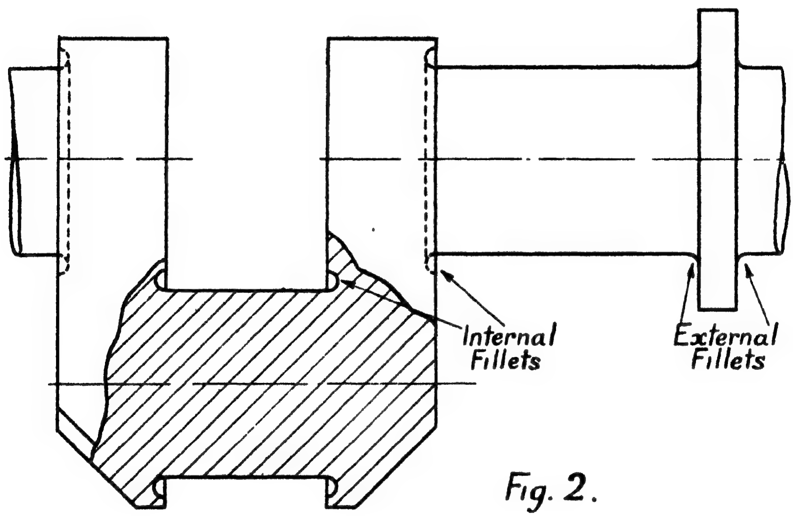



Fig. 2.

Fig. 2 shows the modification possible and is taken from modern practice in oil-engine design.

Overstressing, i.e. the deliberate stressing of material beyond its elastic limit, is often adopted with safety. Crane hooks of standard design, for example, are overstressed under the proof load.†

## 2. Cheap Materials and Methods of Fabrication.

These are interlinked: for example, cast iron, which is cheap, is only acceptable if the operation of casting results in the cheapest satisfactory article. Actually, the dearer material, steel, in plates and sections fabricated by welding, is being widely used in lieu of many castings because the finished product is cheaper. Expensive machine tool work on castings can often be avoided by the use of welded steelwork. The designer must therefore acquaint himself with the general technique and relative cost of production, not only by simple casting, forging and machining, but by welding, stamping, extruding and die casting. The student might with advantage investigate the possibility of using welded designs for many of the machine parts described in this book; in this he should bear in mind two points: (*a*) that a welded job is usually only worth while if for the same strength it is lighter than the corresponding forging or casting; (*b*) that bending, for plates, is cheaper than welding.

* "The Relation of Fatigue to Modern Engine Design", by Burn, *The Engineer*, 5. 7. 1935.
† "Design of Crane Hooks", *Proc. I. Mech. E.*, 1934.

**3. Hand Processes.**

The elimination of hand processes usually results in a cheaper and a better engineering job. By applying the principles of kinematic design in engineering, much hand fitting and maladjustment may be avoided. An important proposition concerning couplings states that if the constraints between two members exceed six, the members will suffer unnecessary elastic deformation. These principles have been exploited in the design of scientific instruments with admirable results. Space does not permit a discussion of kinematic principles here, and the student should refer not only to standard textbooks but particularly to the reference given.*

**4. Planning for Production.**

For commercial success in the production of an article on a repetitive basis, e.g. a motor-car engine, it is just as important that it be manufactured in the most economical way as that it be scientifically designed. The designer must not only settle its proportions for the specified duty, but he must incorporate features which will tend towards cheapening the cost of production: *he must plan its production.*

A study of the commoner types of jigs and fixtures used to facilitate machining operations is obviously necessary before special provision can be made for their application in the design of a machine part. But apart from the study of the advanced technique involved here, the student can constantly seek to incorporate in his drawings features that will cheapen production. For example: holes, all of uniform diameter, and lying on a circle, will obviate drill changes and permit the use of a rotating jig—compare fig. 1*b* with fig. 1*a*; seatings giving three-point rather than area contact make easier fitting—compare fig. 2*b* with fig. 2*a*; facings brought out from their normal position make machining simpler, i.e. (*b*) and (*c*) are preferable to (*a*), fig. 3; and so on. Other examples will be recognized in practice: small connecting rods are finished with the rod and cap in one piece, being afterwards separated, for cheapness; gear wheels are commonly designed so that all teeth may be finish ground; a bearing of the type shown in fig. 4 is better if arranged as at (*b*) rather than (*a*), on account of the difficulty of gauging the internal bore.

The application of welding processes, in conjunction with the use of suitable jigs and fixtures, offers much scope for cheap and reliable production: e.g. in the manufacture of very large stators for electric generators, this method of fabrication has been employed with most satisfactory results and the complete elimination of all major machining operations.

The keen student will enjoy making a revision of his drawings with points such as the above in mind.

**5 and 6. Standard Parts: Interchangeability.**

Large numbers of components are now standardized. The designer must familiarize himself with the specifications of the British Standards Institution and use standard components wherever possible. Further, other parts must be completely specified on the working drawing as regards finish and limits of accuracy. The settlement of limits and tolerances is discussed on the pages which follow. This section is extremely important; and the student should always question the adequacy of a drawing on which dimensions are given without tolerances.

## EXERCISES

(1) A casting of symmetrical design is shown in fig. 5: the principal dimensions are given for general guidance only. Sketch a design in welded steel, which will be equally rigid and satisfactory, to replace the casting. Use bent material wherever possible.

(2) The brackets shown on pp. 168, 169 and 202 lend themselves to fabrication by welding. Prepare alternative designs, adopting the best construction for the purpose of fabrication by welding.

* "Kinematic Design in Engineering", Prof. Pollard, *Proc. I. Mech. E.*, 1933.

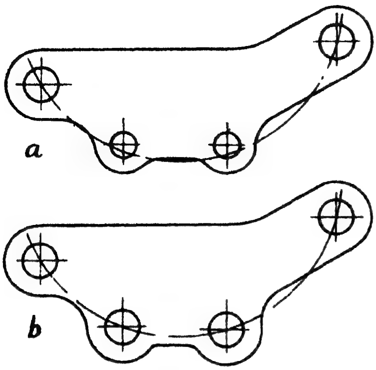



Fig I.

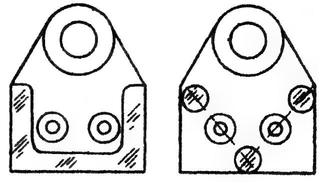



Fig 2

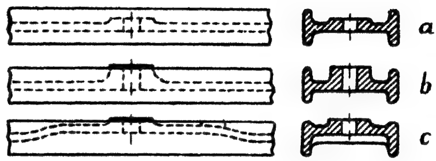



Fig 3.

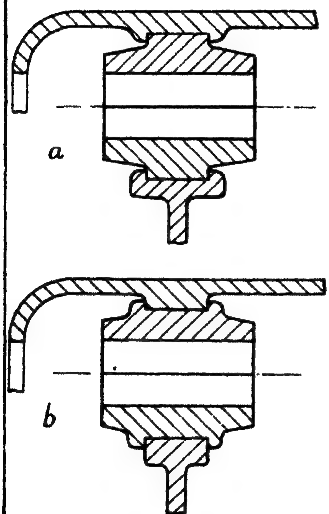



Fig 4.

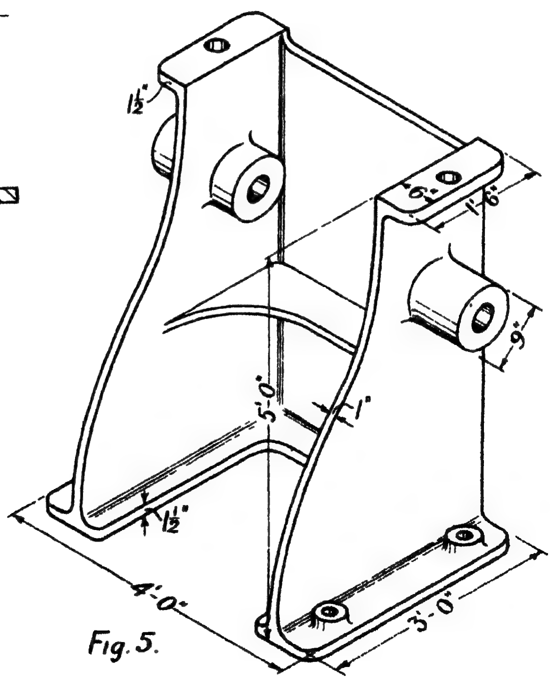



Fig. 5.

The dimensions given in the various drawings throughout this book have been shown usually as integers and fractions, e.g. $2\frac{3}{16}''$, $10\frac{1}{8}''$, and not often as decimals. This has been done for the sake of simplicity and clearness; and although drawings of engineering parts are still dimensioned in this way, there are many others for which something more is required—and it is this requirement which will be discussed here.

**Interchangeability.**—Where an engineering article is being produced in quantity, it is usually necessary to ensure that the mating parts, taken at random, can be assembled without further fitting and machining. This requirement applies also to replacements. As an example, a new connecting rod for the engine of a motor car must give the same clearances and fits as the one it replaces; and a replace engine must fit into the position of the old one.

To ensure this interchangeability it becomes necessary to specify the dimensions of the parts closely by means of **limits**. As an example, a dimension of 2″ may be shown as $\frac{2{\cdot}005}{1{\cdot}995}$, indicating that the actual size must lie between these extreme limits. The difference between such high and low limits of size is called the **tolerance,** which is the permissible variation selected as being sufficient to cover all reasonable imperfections of workmanship.

The use of limits is a step to be taken with a full appreciation of the consequences; for tolerances may involve the use of jigs, and hence may increase the manufacturing time and cost of an article if few of it are required. In quantity production, however, the additional cost of jigs to give fine tolerances has to be balanced against the time and money saved by straightforward assembly (or replacement during service). The assessment is a complex one involving many factors. Drawings of even high-grade components which are required in small quantities are often given without limits because it is known that a certain grade of accuracy will be attained normally in the Firm's workshops, the "run-of-shop" accuracy.

To illustrate the point, holes drilled (without jigs) in metal are found usually to be slightly larger than the drill, because of the difficulty of grinding the cutting end centrally; further, their centres will usually not coincide exactly with the theoretical positions, because the drill will wander slightly. By inspection and measurement it is possible to determine the extreme divergences which obtain in the shops for holes of various sizes, and over a period of manufacture. In well-equipped shops and with skilled workmen, the sizes of small drilled holes up to $\frac{1}{4}''$ diameter may be larger by amounts not exceeding ·005″; and the sizes of 1″ holes by amounts not greater than ·020″; similarly, their positions will deviate from the required positions by amounts up to but not exceeding ·010″. If, therefore, such errors are within the limits of accuracy permissible for the component being manufactured, then there may be no need to give limiting dimensions; but if quantity production and interchangeability are required, then full and proper tolerancing is necessary.

A decision to introduce limits to a dimension, and the settlement of the tolerances to be adopted for correct functioning, require from the designer not only a knowledge of the results of successful practice but familiarity with workshop processes and methods. Consider, for example, the pair of flanges shown in fig. 1. By providing limiting dimensions for the bolt holes and the coupling bolts, it would not be impos-

sible to ensure that any two flanges and any bolts could be assembled with the certainty that the bolts when inserted consecutively would be push fits in the holes. To achieve this, however, the tolerances would have to be so small as to make the cost of manufacture quite prohibitive; and the

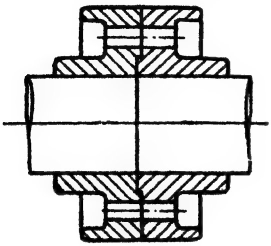


Fig. 1

usual practice is to leave the flange holes small and to use a reamer to give the finished sizes after the coupling halves have been assembled on the shaft. Many similar considerations apply when limiting dimensions are to be provided.

**Limiting Dimensions: Conventional Practice.**—It is recommended that limiting dimensions should be shown with the maximum and minimum dimensions above and below the dimension line, thus $\underset{2{\cdot}998}{\xleftrightarrow{3{\cdot}003}}$, with the larger dimension above the line.

It is often the practice to show a **basic** dimension, together with a tolerance expressed in decimals; and the tolerance may be given on one side of a basic dimension (**unilateral**), or on each side (**bilateral**), thus:

Unilateral $2{\cdot}998\,{}^{+\,{\cdot}005}_{-\,0}$; Bilateral $3\,{}^{+\,{\cdot}003}_{-\,{\cdot}002}$.

For working drawings, the limits should be preferably shown thus: $\frac{3{\cdot}003}{2{\cdot}998}$, rather than $2{\cdot}998\,{}^{+\,{\cdot}005}_{-\,0}$, so that the workman is not involved in any calculation.

The letters H and L, standing for "high" and "low", are often used to prefix limiting dimensions, thus: $\begin{matrix}\text{H. } 3{\cdot}003 \\ \text{L. } 2{\cdot}998\end{matrix}$. The letters are superfluous unless their insertion carries some additional information. In some systems the use of H and L means that in no circumstances must the actual dimensions fall outside these limits; whereas the absence of H and L indicates only that the grade of fit must be adhered to and that a slight variation in basic size is permissible—a variation which obviously would not permit universal interchangeability.

**Holes and Shafts.**—In most engineering products there are both plain and screwed cylindrical parts which have to fit one within the other, e.g. a shaft within a hole, with appropriate tolerances for correct functional operation; these cases are perhaps the commonest for the application of limiting dimensions. Plain cylindrical forms only will be considered here, and in what follows reference has been made freely to British Standard 164, to which the student should later refer. (Limits for B.S. Screw Threads are dealt with in B.S. 84.)

The conventional method of showing tolerance **zones** for a hole and shaft are given in fig. 2, page 162 which is self-explanatory. A dimension of particular importance is that called the **allowance.** This is a prescribed difference between the high limit for the shaft and the low limit for the hole; it may be positive or negative according as the shaft is to be free or fixed in the hole.

The smallest hole and largest shaft condition, often referred to as the "maximum metal" condition, is a vital factor for interchangeability. It

will be seen that: maximum clearance − minimum clearance (or allowance) = tolerance on shaft + tolerance on hole. The design procedure is to settle the maximum and minimum clearances, and to apportion the aggregate tolerance thus given between the mating components. The magnitude of these tolerances must then be compared with the known performances of the machine tools to be used for the work and against the accuracy required in the manufacture of the gauges for inspecting the work.

The kind of fit between a shaft and a hole may be shown in various ways. In one, the hole may be regarded as being constant in diameter while the various fits are obtained by changing the diameter of the shaft, as in fig. 3. This method, called the **Hole Basis,** is that used in the construction of the British Standard Tables. Conversely, the shaft may be regarded as the constant member and the hole made the variant; this is called the Shaft Basis, and is not recommended as a British Standard—although it sometimes has to be used.

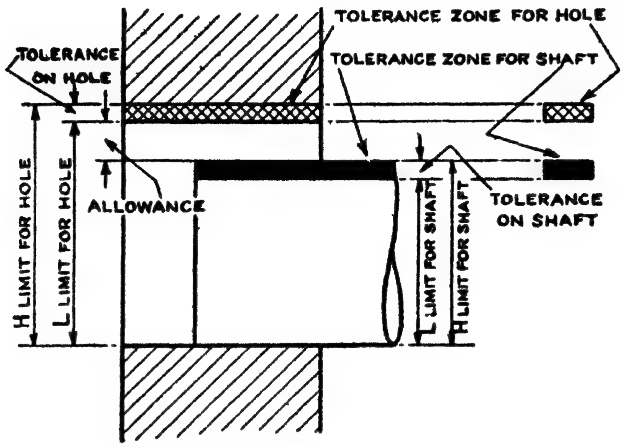



Fig. 2

HOLE BASIS

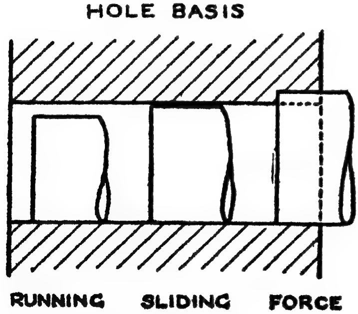



Fig. 3

The application of the **Unilateral System** of dimensioning with the **Hole Basis,** which is the recommended combination, requires that the lower limit of the hole becomes the basic size of the hole.

For a normal running shaft in a $1\frac{1}{4}''$ hole, suitable limiting dimensions would be:

Hole: $\begin{matrix} 1{\cdot}2512 \\ 1{\cdot}25 \end{matrix}$ or $1{\cdot}25 \begin{matrix} + \cdot 0012 \\ - 0 \end{matrix}$

Shaft $\begin{matrix} 1{\cdot}2482 \\ 1{\cdot}247 \end{matrix}$ or $1{\cdot}2482 \begin{matrix} + 0 \\ - \cdot 0012. \end{matrix}$

The low limit for the hole is the basic dimension, $1{\cdot}25''$. The high limit

for the shaft is 1·2482, so that the **allowance** is ·0018; this represents the minimum clearance. The maximum clearance may evidently be as much as 1·2512 − 1·247 = ·0042.

### Selection of Limits for Holes and Shafts.

**Holes.**—It has been stated that only the unilateral system, with the hole basis, is recommended as British Standard practice. The hole basis implies that the hole is to be constant in basic size and that any variation in "allowance" is to be made on the shaft. This does not mean, however, that for all classes of work a hole with a given basic diameter is to have a certain universal tolerance. The B.S. table of Limits for Holes, an extract of which is given above (fig. 4), shows four standards for hole tolerances, described by the letters B, U, V and W.

HOLES

H = High limit of tolerance.
L = Low limit of tolerance.

TOLERANCE UNIT = 0·001 INCH.

| Nominal Sizes. | UNILATERAL HOLES. (In which the Low Limit of the Hole is Nominal Size.) | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | B | | U | | V | | W | |
| Inches. | H | L | H | L | H | L | H | L |
| 0 to 0·29 | +0·3 | 0 | +0·6 | 0 | +1·2 | 0 | +2·4 | 0 |
| 0·3 to 0·59 | +0·4 | 0 | +0·8 | 0 | +1·6 | 0 | +3·2 | 0 |
| 0·6 to 0·99 | +0·5 | 0 | +1·0 | 0 | +2·0 | 0 | +4·0 | 0 |
| 1·0 to 1·49 | +0·6 | 0 | +1·2 | 0 | +2·4 | 0 | +4·8 | 0 |

Fig. 4

Table B gives tolerances for work of a precise nature; table U is for general engineering practice; and tables V and W give more liberal tolerances for rougher engineering work.

As an example, for a 1″ hole, the tolerances would be:

| | | |
|---|---|---|
| Table B | .. .. | +·0006 |
| Table U | .. .. | +·0012 |
| Table V | .. .. | +·0024 |
| Table W | .. .. | +·0048 |

Only Table U will be considered further here.

**Shafts.**—The basic diameter of a shaft for a given hole will be varied to give the kind of fit required, which may range from clearance to interference. In B.S. 164, 14 classes of shaft dimensions are standardized, and these, taken with a U hole (see fig. 4), will give the following classes of fits, from a "heavy drive" to a "coarse clearance" (fig. 5).

| Designation. | Description of fit. | Class of fit. |
|---|---|---|
| UF | Heavy drive | Interference |
| UE | Light drive | |
| UD | Heavy keying | Transition |
| UC | Medium keying | |
| UB | Light keying | |
| UK | Push | |
| UL | Slide or Easy push | Clearance. |
| UP | Easy slide or Close running | |
| UM | Close running (1) | |
| UQ | Close running (2) | |
| UR | Normal running | |
| US | Slack running | |
| UT | Extra slack running | |
| UTT | Coarse clearance | |

Fig. 5

## SHAFTS

H = High limit of tolerance.
L = Low limit of tolerance.

TOLERANCE UNIT = 0·001 INCH.

| Nominal Sizes. | F | | E | | D | | C | | B | | K | | L | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Inches. | H | L | H | L | H | L | H | L | H | L | H | L | H | L |
| 0 to 0·29 | +1·2 | +0·9 | +0·9 | +0·6 | +0·6 | +0·3 | +0·4 | +0·1 | +0·3 | 0 | +0·1 | −0·2 | 0 | −0·3 |
| 0·3 to 0·59 | +1·6 | +1·2 | +1·2 | +0·8 | +0·8 | +0·4 | +0·6 | +0·2 | +0·4 | 0 | +0·2 | −0·2 | 0 | −0·4 |
| 0·6 to 0·99 | +2·0 | +1·5 | +1·5 | +1·0 | +1·0 | +0·5 | +0·7 | +0·2 | +0·5 | 0 | +0·2 | −0·3 | 0 | −0·5 |
| 1·0 to 1·49 | +2·4 | +1·8 | +1·8 | +1·2 | +1·2 | +0·6 | +0·9 | +0·3 | +0·6 | 0 | +0·3 | −0·3 | 0 | −0·6 |

| Nominal Sizes. | P | | M | | Q | | R | | S | | T | | TT | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Inches | H | L | H | L | H | L | H | L | H | L | H | L | H | L |
| 0 to 0·29 | −0·2 | −0·5 | −0·3 | −0·6 | −0·5 | −0·9 | −0·9 | −1·5 | −1·5 | −2·4 | −2·4 | −3·6 | −3·6 | −6·0 |
| 0·3 to 0·59 | −0·2 | −0·6 | −0·4 | −0·8 | −0·6 | −1·2 | −1·2 | −2·0 | −2·0 | −3·2 | −3·2 | −4·8 | −4·8 | −8·0 |
| 0·6 to 0·99 | −0·3 | −0·8 | −0·5 | −1·0 | −0·8 | −1·5 | −1·5 | −2·5 | −2·5 | −4·0 | −4·0 | −6·0 | −6·0 | −10·0 |
| 1·0 to 1·49 | −0·3 | −0·9 | −0·6 | −1·2 | −0·9 | −1·8 | −1·8 | −3·0 | −3·0 | −4·8 | −4·8 | −7·2 | −7·2 | −12·0 |

Fig. 6

**Illustrations, for Holes and Shafts.**

The table (fig. 6) gives extracts from B.S. 164 (Limits for Holes and Shafts) for shafts up to 1·49″ diameter. The full table goes up to diameters of 25·29″.

The student would do well to investigate the variation in allowances which the use of these tables will give.

Let us consider a 1″ hole. From the U table the tolerance is +·0012.

The diameter of a "heavy drive" shaft, table F, will lie between 1·0024 and 1·0018. Hence the allowance, or maximum interference (between H

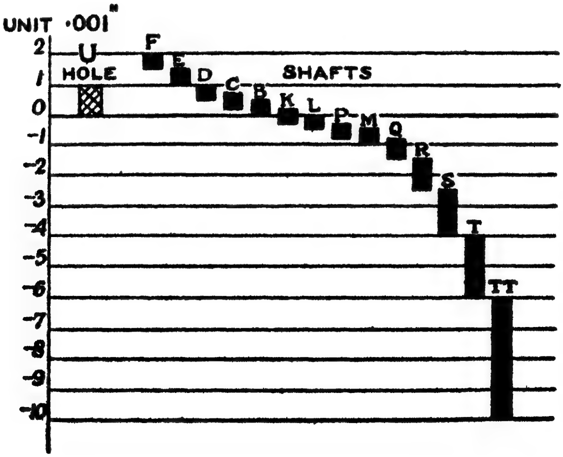



Fig. 7

limit for shaft and L limit for hole), is ·0024; the minimum interference is ·0006.

Similarly, the diameter of an "easy slide" shaft, table P, will lie between ·9997 and ·9991. Hence the "allowance", or minimum clearance, is ·0003; the maximum clearance is ·0021.

The allowances for each class of fit should be similarly deduced and given careful consideration. If, for the sliding fits, the student can actually handle examples of shafts and holes, he will gain a better appreciation of the effect of the tolerances given.

The tolerance zones can be well illustrated graphically by a chart such as that shown in fig. 7, which has been prepared for a range of sizes from 0·6" to 0·99", from fig. 6.

**Departure from Tolerance Tables.**—It is often necessary to give special limits, e.g. smaller tolerances for larger shafts, than those given in the standard tables. This may be done by taking values from adjacent columns which will give the tolerances required. Some engineering firms find the B Hole tolerances too fine for their requirements and adopt something lying between the U and the B figures. Again, other firms do not work to closer differences between limits than ·0002, and they "round up" the B.S. recommended figures.

**Circularity and Parallelism.**—Unless otherwise specified, it has to be assumed that the limiting dimensions for shafts and holes cover the requirements for circularity and parallelism. For example, the U tolerance on a 12" diameter hole is +·0034". If it is permissible for a cross-section to be oval, or irregularly shaped, then providing its outline lies wholly between two concentric circles of 12" and 12·0034" diameter, or if longitudinal inaccuracies are unimportant providing they lie between two concentric cylinders of these diameters, no other limiting dimensions are required. There may be cases, however, where circularity and parallelism are vitally important and where close limits must be imposed. The designer has always to keep in mind the fact that very fine tolerances for circularity or parallelism involve problems in gauge manufacture and in measurement which cannot be considered apart, or relegated to the jig and tool and inspection departments.

**Eccentricity.**—If a shaft is stepped, and if it is important that its parts should be coaxial to fine limits, then it will be necessary to define the permissible eccentricity of one cylinder relative to the other (fig. 8).

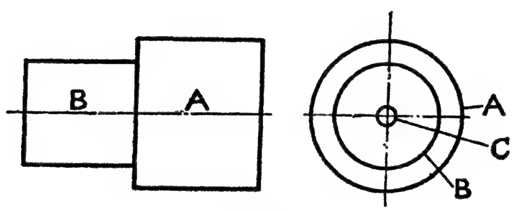



Fig. 8

For example, if A is chosen as the datum cylinder, and if an eccentricity tolerance of ·001" is required for the cylinder B, then B may have its axis anywhere on the surface of a small cylinder C of radius ·001" (or, in an end view, B may have its centre anywhere on circle C of radius ·001").

The shaft diameters will also have limits assigned to them; and the student should pursue the implications of assigning an eccentricity tolerance in combination with the usual size tolerances.

In practice designers avoid, if they can, double fit parts involving the concentricity of cylinders.

**The Dimensioning of Lengths.**—In the dimensioning of lengths care must be taken not to duplicate tolerances. Decisions must be taken as to which lengths require limiting dimensions (fig. 9).

For example, in the simple part shown, separate dimensions A, B and

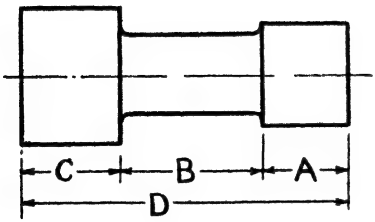



Fig. 9

C would, if in round figures, often be summarized by the insertion of D, to give the workman the overall size of the piece. This is not permissible if tolerances are inserted; then, one of the four dimensions is superfluous. It must be decided, therefore, which of the lengths are sufficiently important from functional considerations to receive tolerances. Suppose each of A, B and C must be held to within ±·003″,

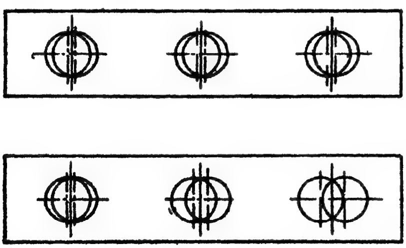


Fig. 10

then clearly D is relatively unimportant and must not be toleranced; its length may, in fact, vary between ±·009″.

**The Tolerancing of Positions.**—Location dimensions have been briefly discussed earlier in the book (see p. 22). Where tolerances are to be assigned, it is vital, in all dimensioning, to select datum or locating surfaces, and to refer all dimensions to these surfaces. The surfaces chosen should be identical, whether the problem is viewed from the functional, dimensional or measurement aspect. Dimensions on companion parts should be given from the same selected datum surface.

Positional dimensioning is sometimes simplified by specifying, on the drawing, the sequence of production operations.

The "chain dimensioning" (with tolerances) of holes may lead to two interpretations: one, in which it is assumed that the tolerances are taken on either side of a geometrical centre line; the other, in which it is assumed that the tolerances apply to the various possible holes and thus become cumulative (fig. 10). To avoid this, each position should be dimensioned from a datum point or surface.

There are at present no British Standard conventions to cover positional tolerancing, and practice is not uniform.

**Preservation of Balance in Tolerancing.**—It will be obvious from what has been said that the designer should use tolerances only where they are essential, and it should be his first concern then to give them **the greatest values** consistent with the correct functioning of the component. Small tolerances will produce interchangeable components, but it may be at too high a cost.

It has been mentioned earlier that the settlement of final tolerances requires a thorough knowledge not only of the operating requirements of the machine part under design, but also of the workshop processes and final gauging necessary for its production. This knowledge, which must be largely

based upon the results of successful practice, is accumulated by, and interchanged between, many industrial concerns; it is the result of experience, and it is not readily acquired during a student's college course.

Having settled the tolerances permissible, then comes the problem, largely geometrical, of their portrayal on a working drawing in such a way as to leave no possible ambiguity in the mind of the producer of the requirements of the designer. A completely dimensioned working drawing ought to mean one thing only.

Unfortunately, a drawing completely dimensioned in this way might contain such a mass of symbols and tolerances as to overwhelm the workman who has to use it. It would be costly to produce, and the assimilation of its full meaning would take up time in the shops and hence again add to the cost. It is necessary, therefore, that the draughtsman should preserve a sense of balance and should know when to stop short of the ideal for the sake of simplicity and cheapness. To this end, close co-operation with the production departments is essential at all times.

**Exercises.**—A great many of the examples in this book can be toleranced on the lines indicated here, and the student should prepare complete working drawings of a few selected components referring to the layout shown on p. 33. It is suggested that tolerances should be provided, to begin with, for the components shown by the drawings on the following pages: 65, 67, 73, 77, 101, 176, 186, 190.

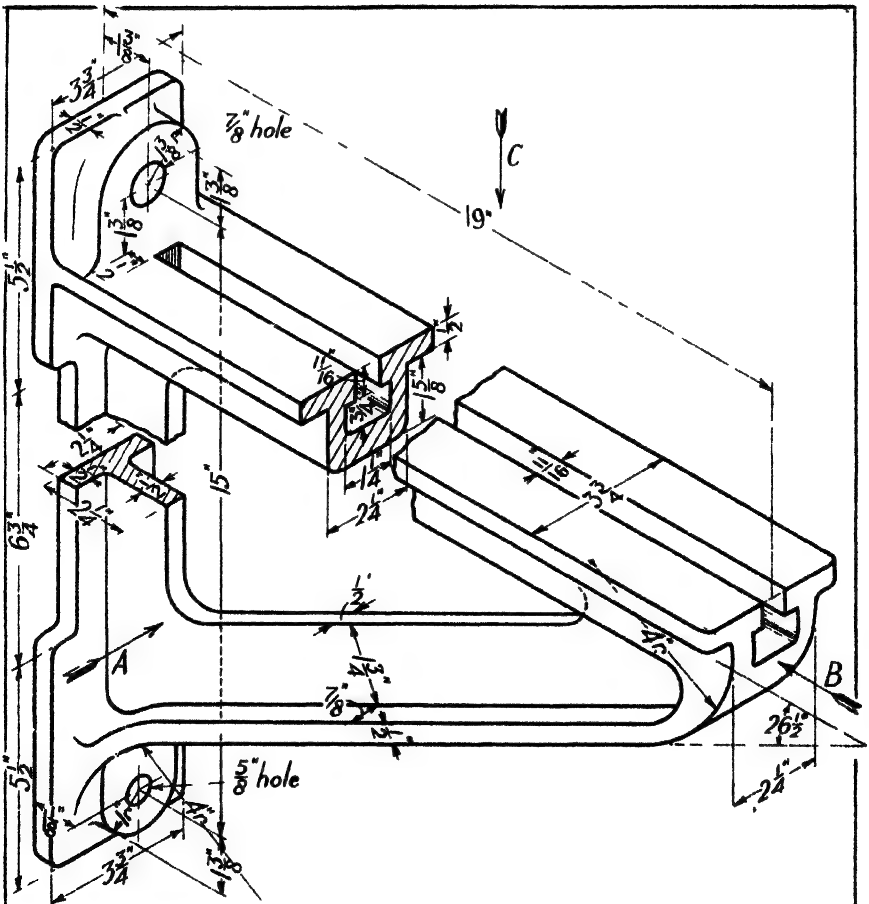
**Note.**—A discussion on Drawing Office Practice in relation to Interchangeable Components appears in the *Proc. I. Mech. E.*, 1945.

# PART II

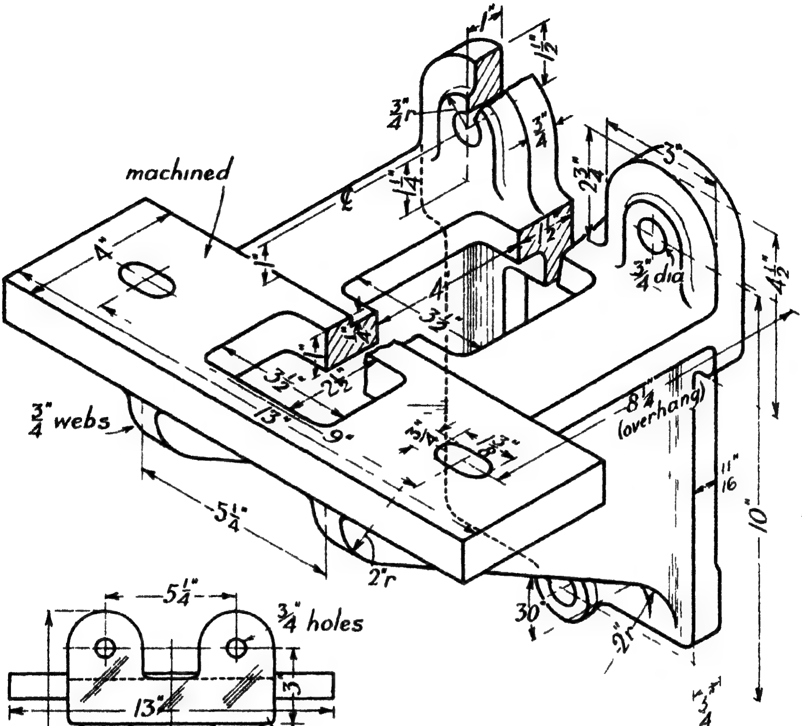
## EXERCISES, SUPPLEMENTARY TO PART I, AND QUESTIONS FROM PAST EXAMINATION PAPERS

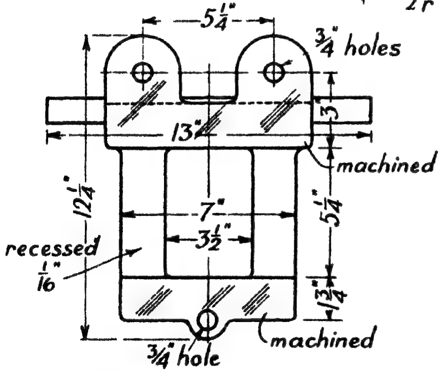
The exercises have been arranged as far as possible in order of increasing difficulty and with some regard to the sequence of the work in Part I The time required to complete each exercise is stated after the question for the general guidance of the student

### CAST-IRON BRACKET



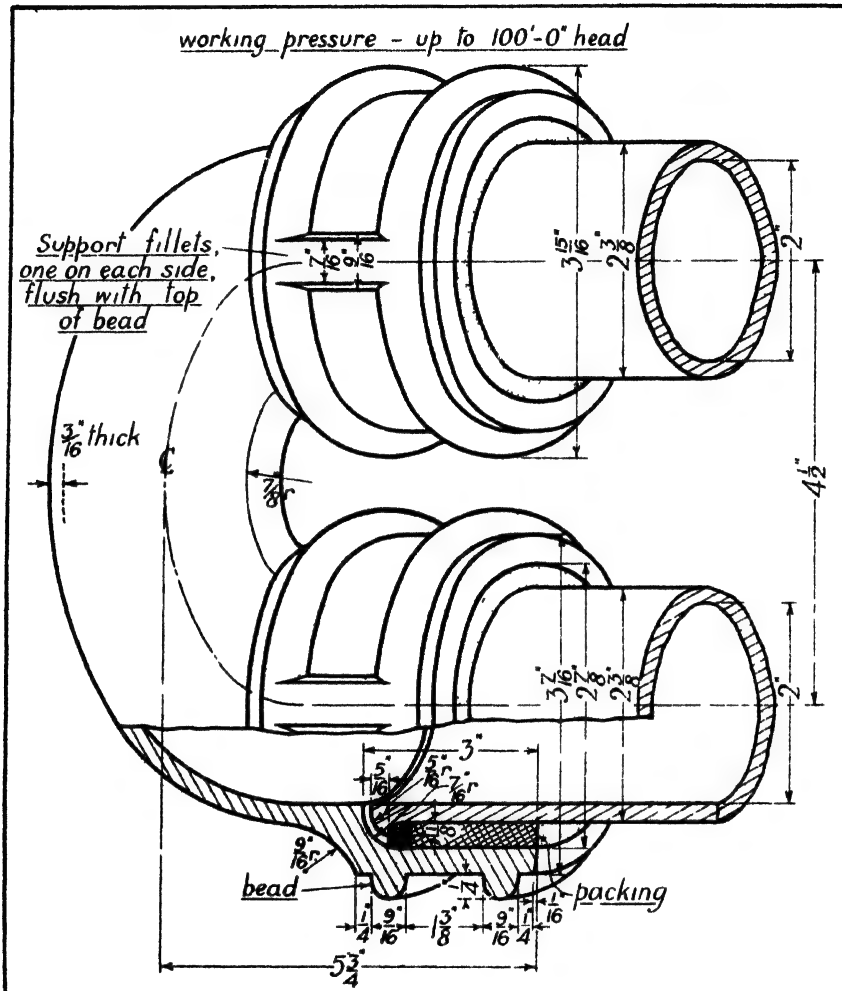

The wall bracket shown is suitable for supporting the pedestals shown on pp 101 and 103 Draw, half size, the following views and dimension them· elevation, in direction A, introducing turned sections, plan, in direction C, end elevation, in direction B. [1½ hr]



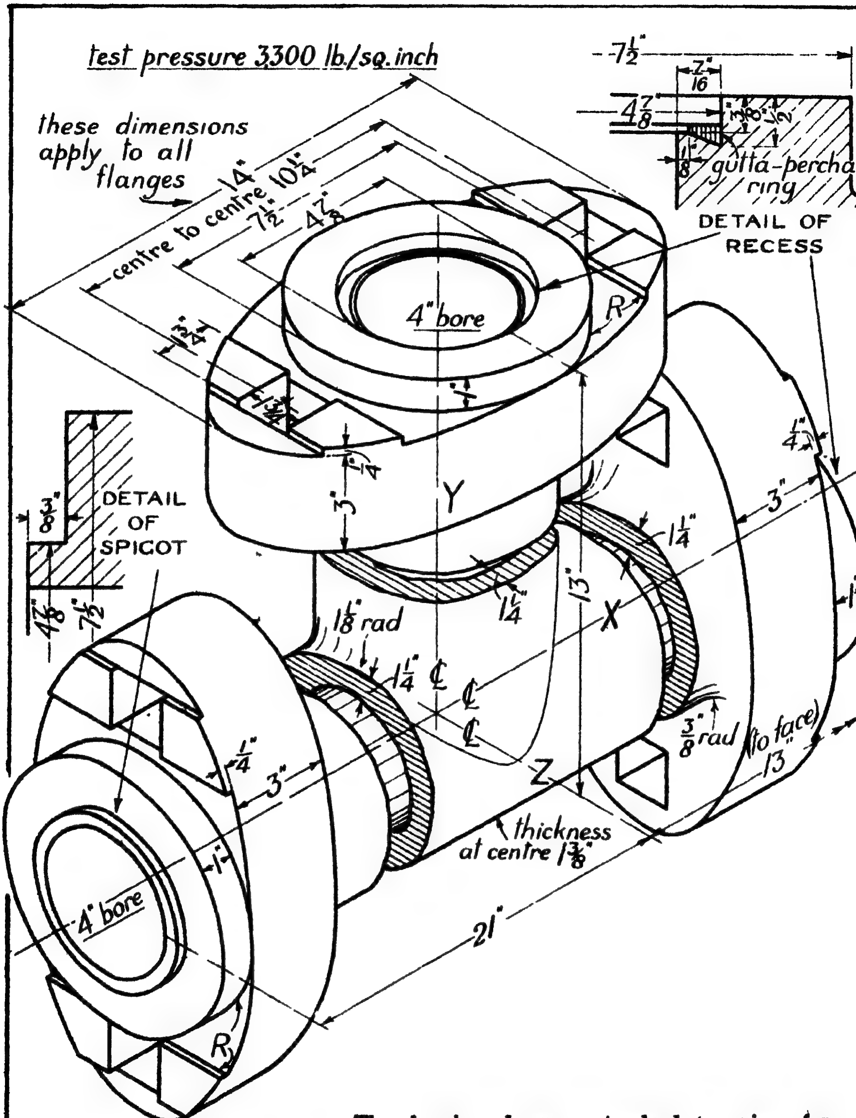

VIEW ON BACK OF BRACKET

**The side wall bracket shown above is also suitable for supporting the pedestals on pp. 101 and 103. The only surfaces to be machined are the seating for the pedestal and the back of the bracket. Prepare, half size, a completely dimensioned working drawing of the bracket, giving the following views: front elevation; end view; plan; sectional end view through the middle. Use third angle projection. Indicate the surfaces to be machined. [2 hr.]**

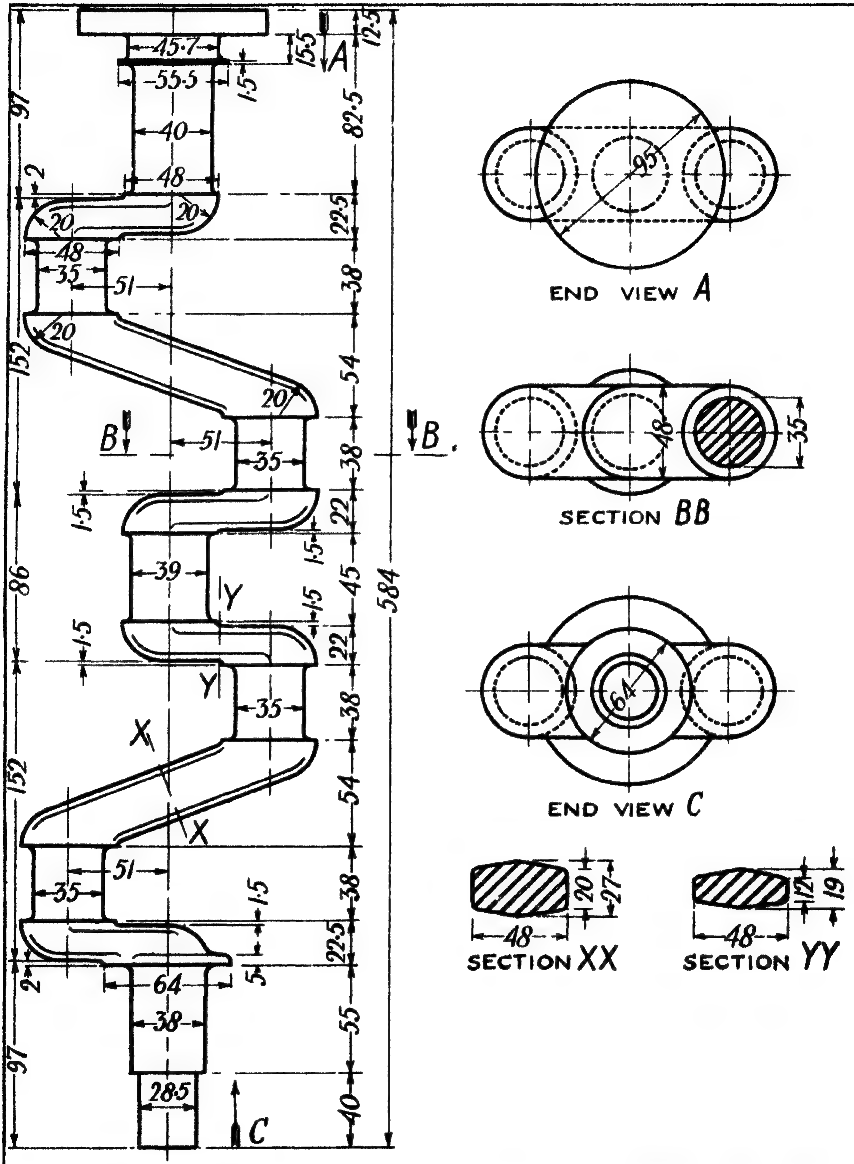



The drawing shows a 2" bore C.I. return bend, with piping, suitable for conveying water at low pressure. The joints for the pipe ends are of the spigot and socket type, the annular space between the pipe and socket being filled with a suitable packing (e.g. gasket and lead).

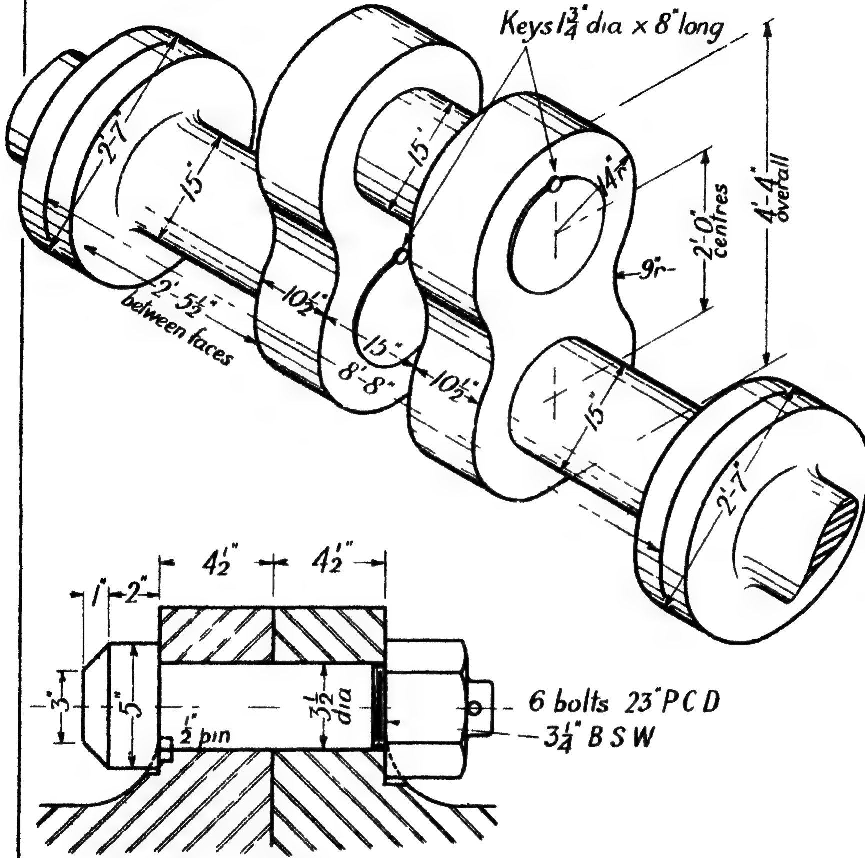
Draw, full size, the following views and dimension them: elevation, showing the upper half of the bend in section; end view from right to left; end view from left to right. Show the pipes broken, as in drawing. [1½ hr.]




The drawing shows a standard tee piece for hydraulic power, working pressure 1200 lb./in.$^2$. The dimensions of the flanges have been taken from Table 9, p. 228. Draw, half size, the following views and dimension them: sectional elevation on a plane containing centre lines X and Y; sectional end view on a plane containing Y and Z; plan. Show the horizontal branch broken to shorten the drawing. Insert small radii at such places as R. [2¼ hr.]



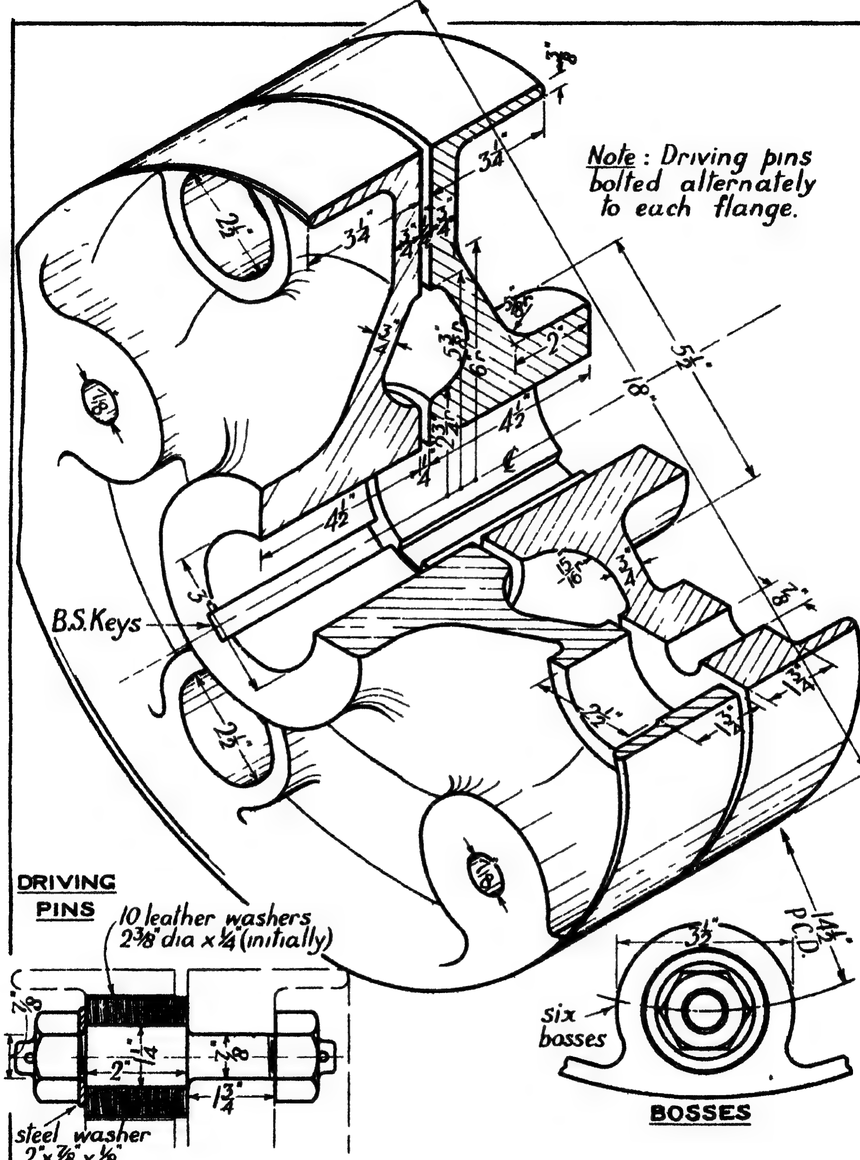

The drawing shows a conventional method of treating irregular crank arms, and provides a test in dimensioning. Draw, half size, the elevation given and project end views from A and C and a view from left to right. Insert dimensions (which are in millimetres). [2 hr.]



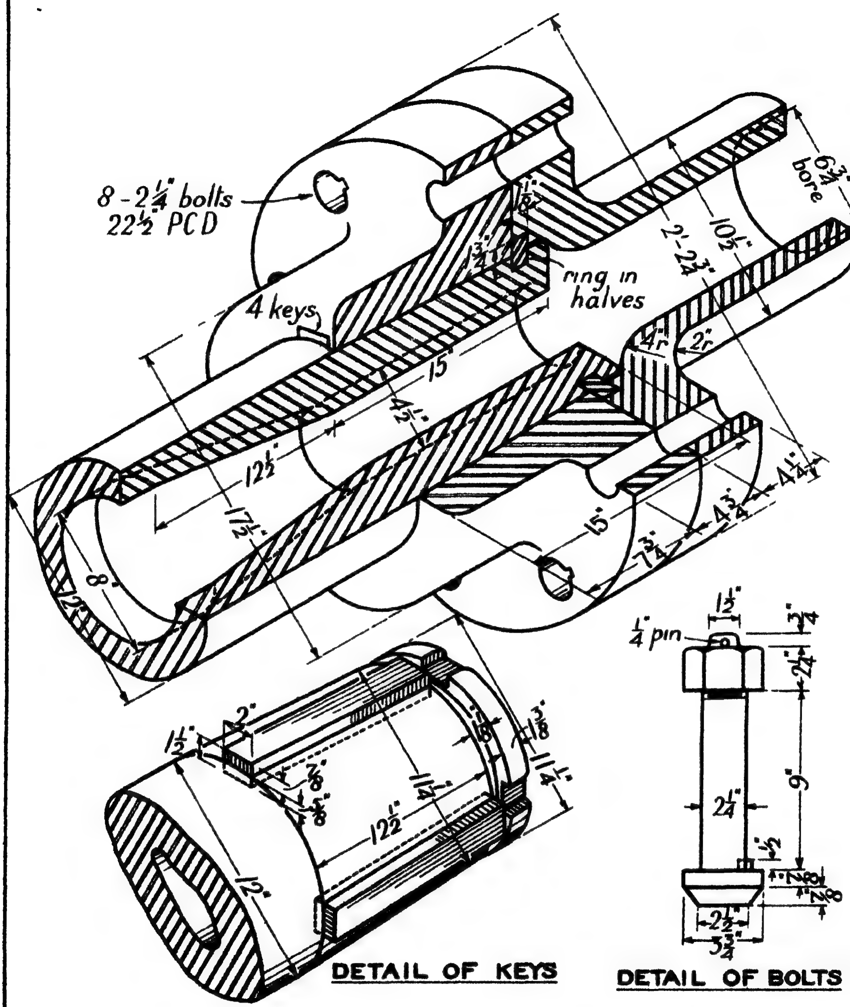

DETAIL OF COUPLING BOLT

The drawing shows one unit of a marine crank shaft of the built-up type. All parts are of steel, machined all over The shafts and pin are shrunk into the arms, round keys being afterwards forced in (or screwed in) for additional security. The complete crank shaft consists of three units as shown, with cranks at 120°.

Draw, to a scale of 1" = 1 foot, an elevation and plan of the unit crank, showing the adjacent shafts attached but broken off. Then project an end view of the complete crank shaft. Dimension the views. Calculate the horse-power of the engine for which the shaft is suitable. Speed 75 revs./min. (see p. 88). *Answer*. 2780 h.p. [2 hr.]

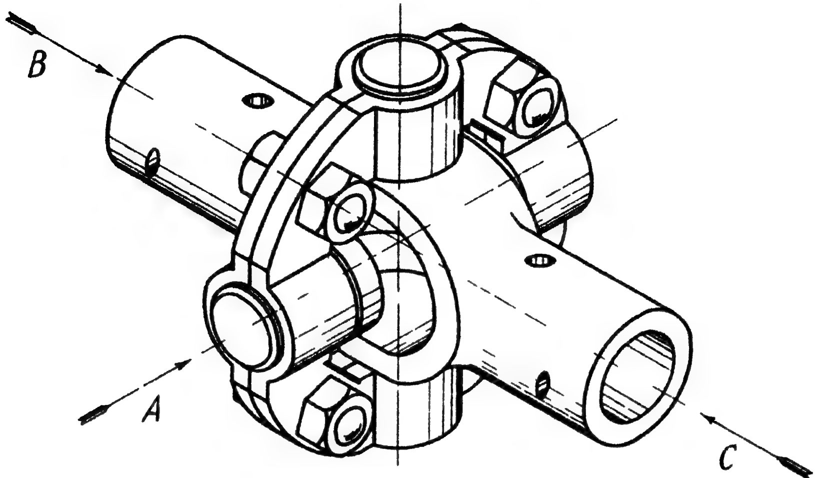



(See also p. 74.) In the coupling shown, power is transmitted from one flange to the other by steel driving pins carrying ring-shaped leather cushions. Draw, half size, the following dimensioned views: sectional elevation of the assembled coupling through one of the pins, and an end elevation. [1½ hr.]

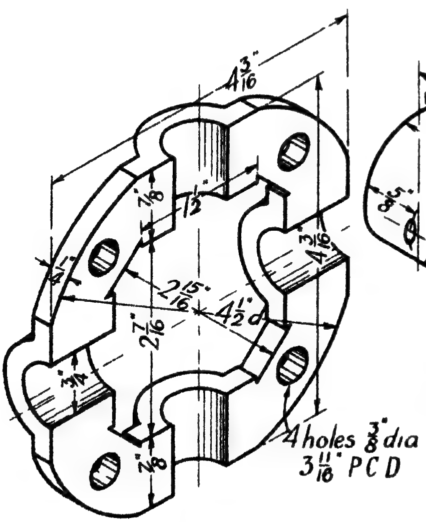



(See also p. 74.) In the coupling shown, torsion is transmitted through four steel keys. The ring prevents the withdrawal of the shaft from the loose coupling. The shafts, keys, ring and bolts are of steel; the loose coupling of wrought iron.

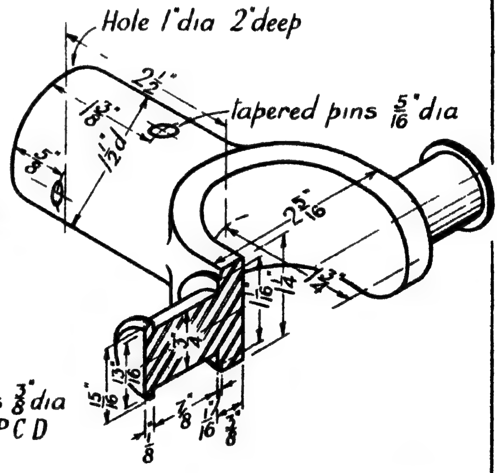
Draw, quarter size, the following dimensioned views: elevation, upper half in section on the shaft axis; end view on the face of the loose coupling. Show a plug screwed into the end of the larger shaft. [2 hr.]




GENERAL ARRANGEMENT

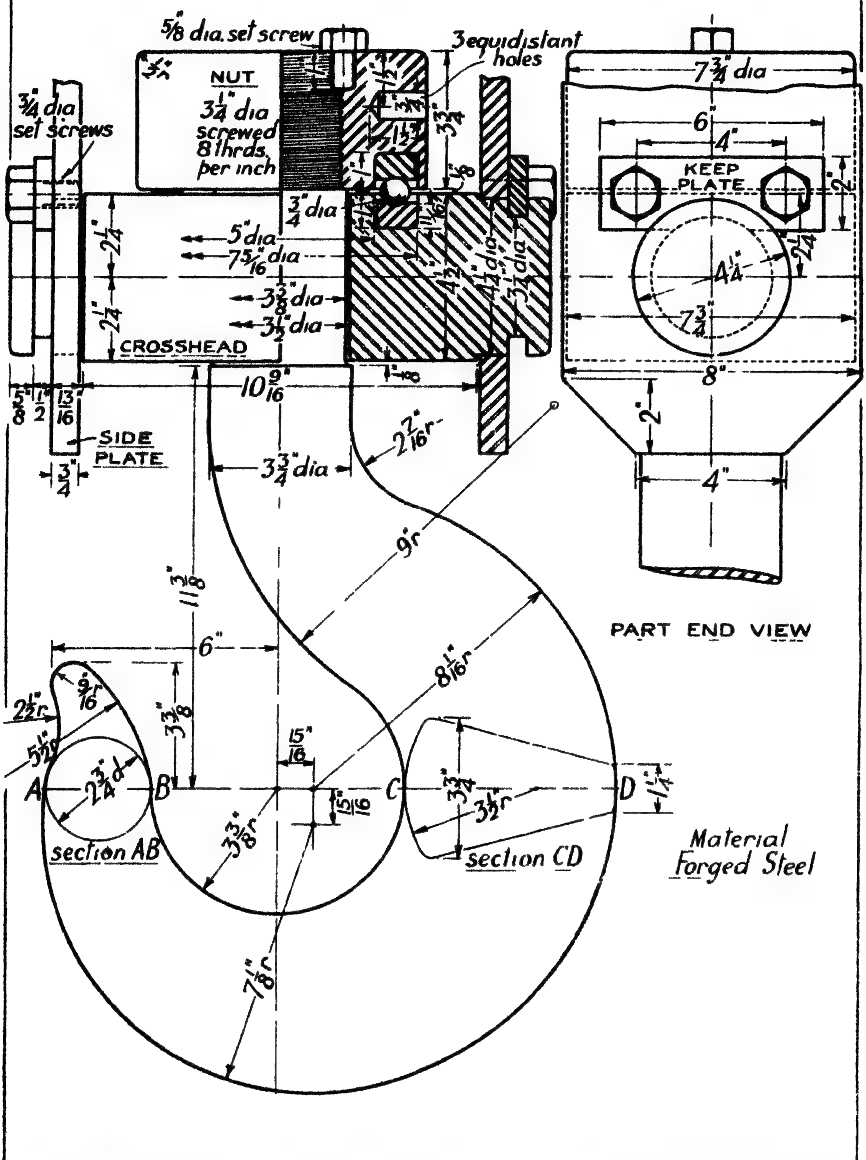



DETAIL OF PLATE

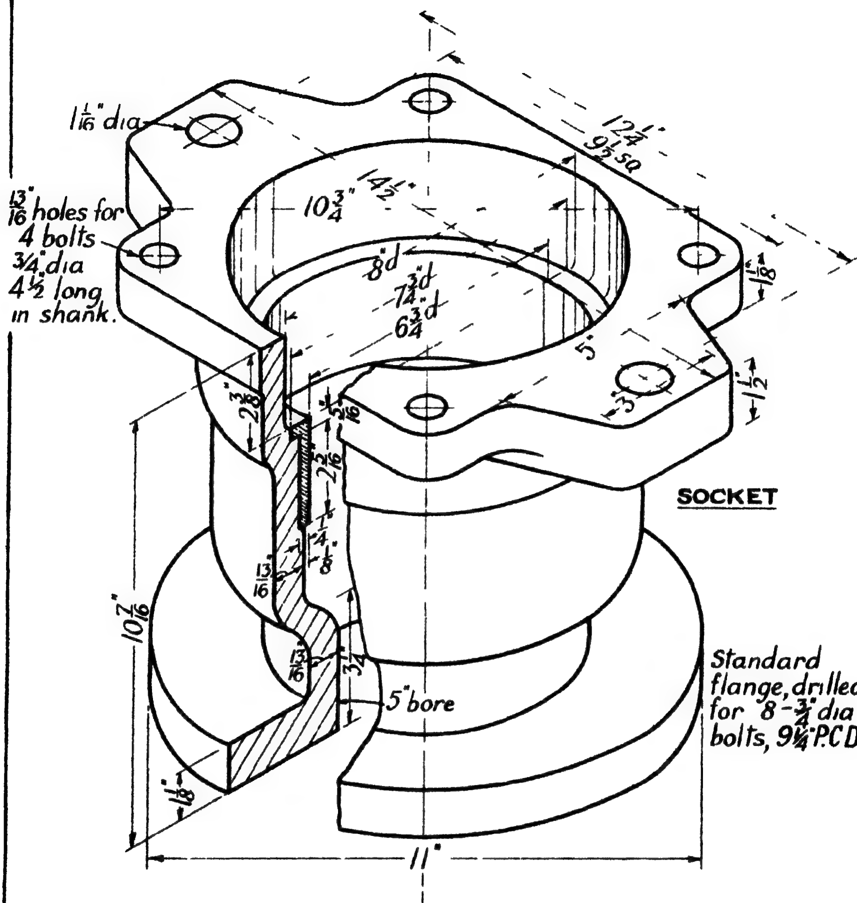



DETAIL OF FORK

**Draw, full size, the following dimensioned views of the assembled coupling: elevation in direction A; end view in direction B with left fork and plate removed, end view in direction C. [2½ hr.]**

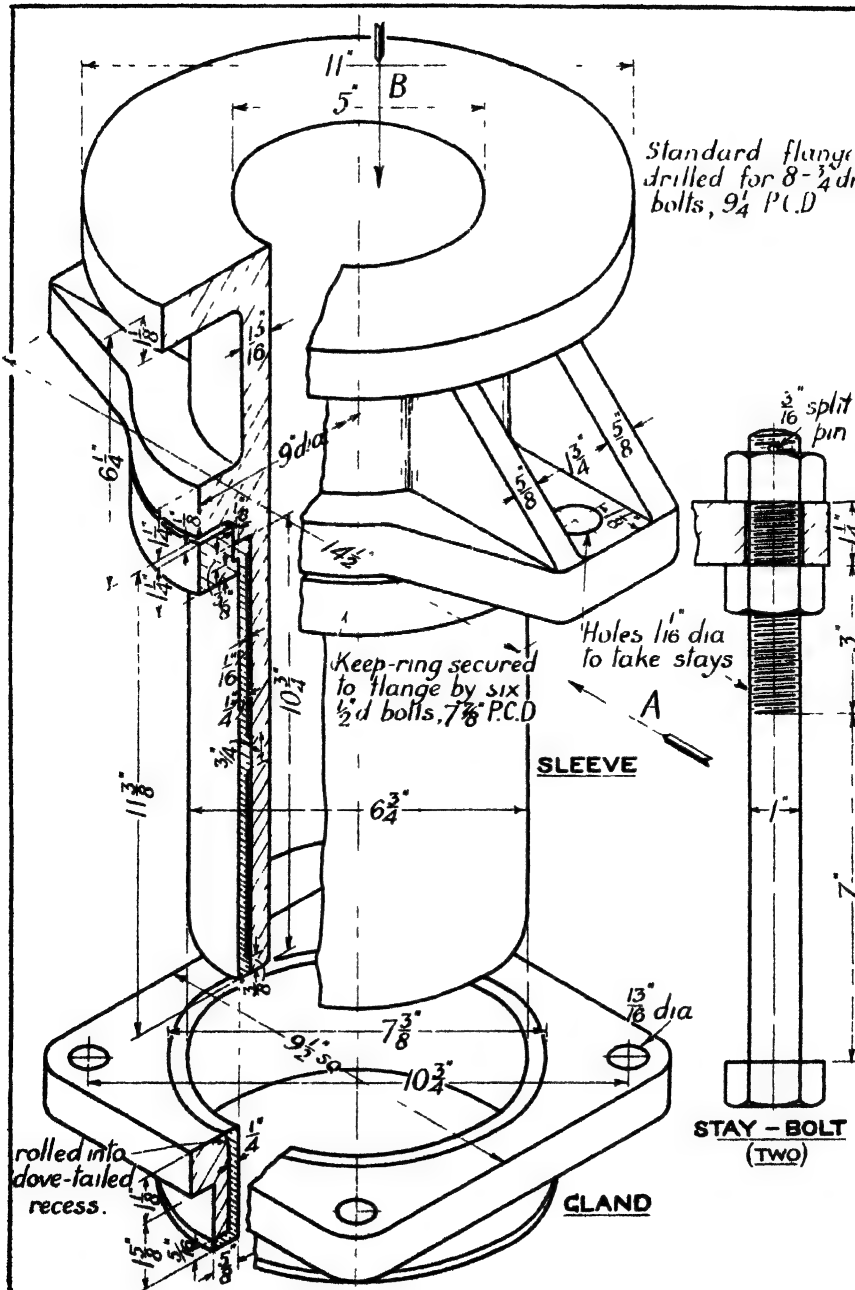


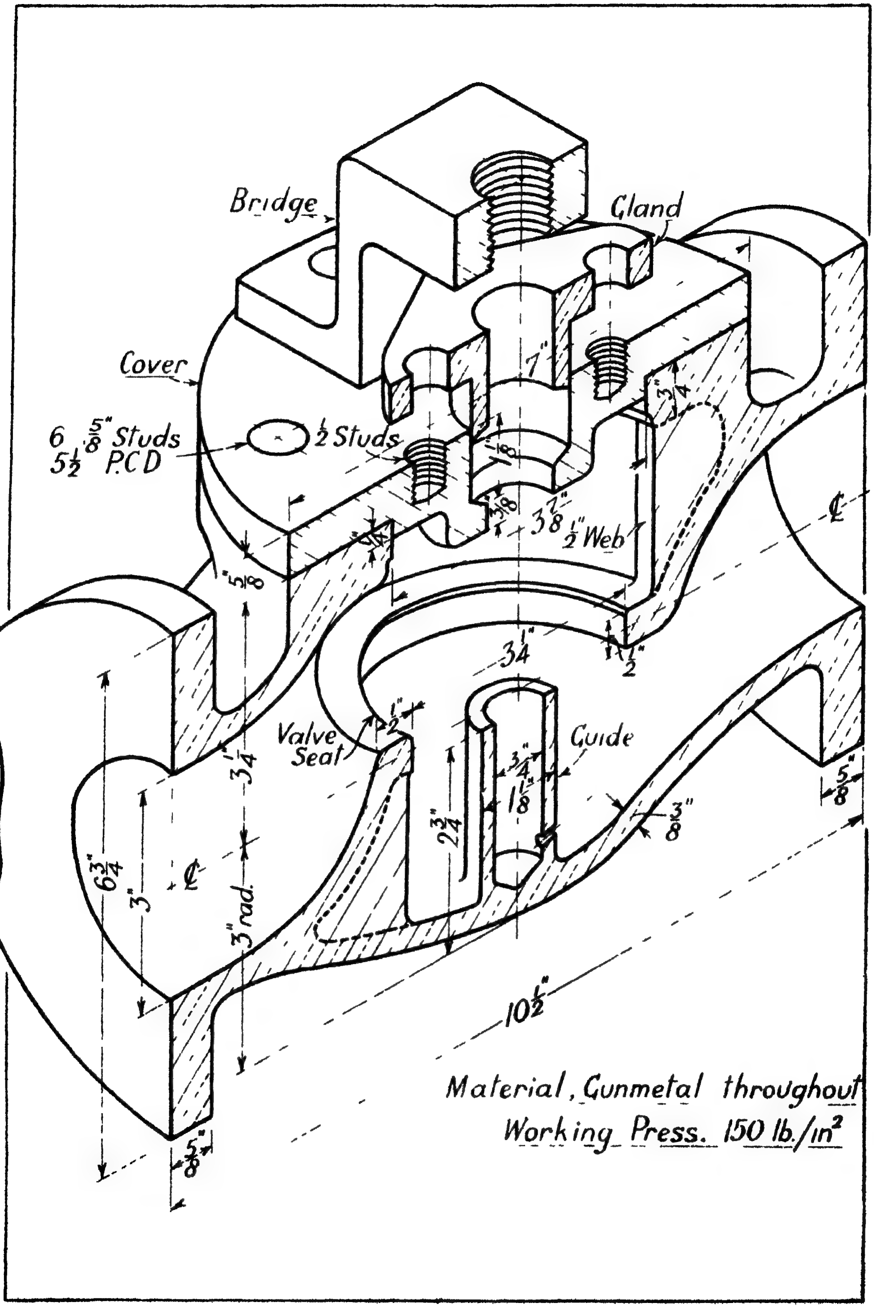

**Draw, half size, the following views of the crane hook, and dimension them: elevation, partly in section, as shown; complete end elevation (from left to right), one half showing the side plate removed to reveal the crosshead. [3 hr.]**

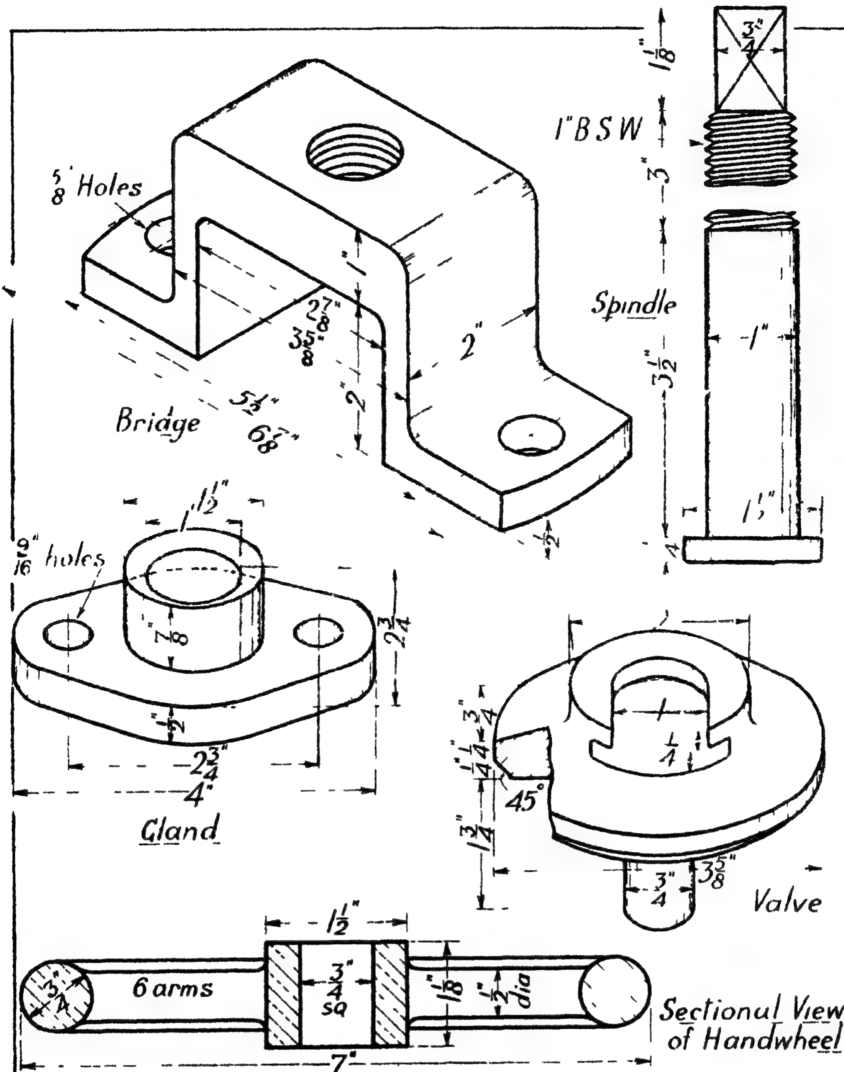



The drawings show the component parts of a C.I. expansion joint for a 5-in. bore steam main, working pressure 150 lb./in.$^2$. The sleeve slides in the socket and the joint is asbestos packed. The complete withdrawal of the sleeve is prevented by stay bolts which pass through holes in the lugs cast on the sleeve and socket. G.M. bushes are fitted to sliding parts in contact: the bush on the sleeve is held in position by a keep ring.

Draw, half size, the following views of the assembled expansion joint with axis horizontal—omit packing and show the sleeve as far into the socket as it will go: elevation in direction A, with upper half in section; plan; end view in direction B, half external and half showing the sleeve in section to reveal the gland. Use third angle projection. Dimension the views and insert finish marks. [5 hr.]


Standard flange drilled for 8-3/4" dia bolts, 9 1/4" P.C.D
Keep-ring secured to flange by six 1/2"d bolts, 7 7/8" P.C.D
Holes 1 1/16" dia to take stays
3/16" split pin
SLEEVE
STAY - BOLT
(TWO)
GLAND
rolled into dove-tailed recess.
13/16" dia
A
B

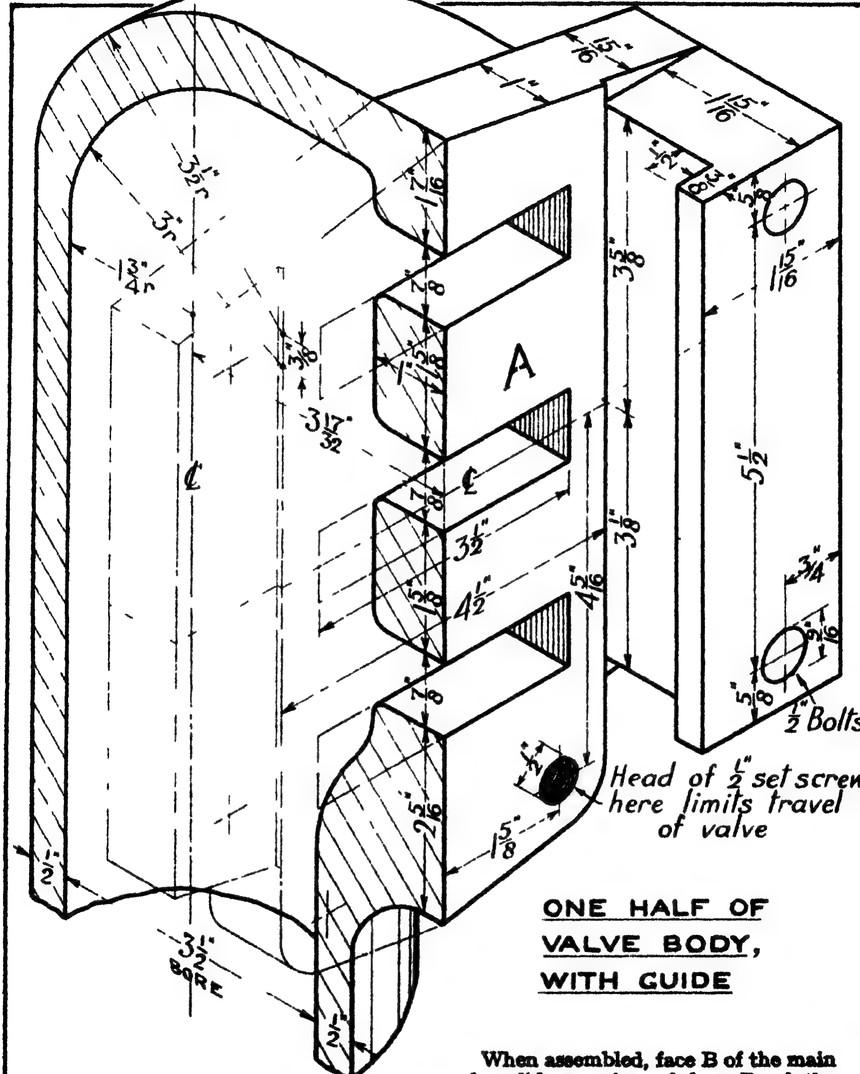

Bridge
Gland
Cover
6 5/8" Studs
5½" P.C.D
½" Studs
½" Web
Valve Seat
Guide
3" rad.
Material, Gunmetal throughout
Working Press. 150 lb./in²




The drawings show the valve box, in section, and the various component parts. The valve seat is in a diaphragm which divides the box and separates the inlet and outlet branches. Transverse sections of the shell of the box are, wherever possible, circular.

Draw, full size, the following views of the completely assembled valve, and dimension them: elevation, half in section; end view, half in section. [4 hr.]

Note.—The settlement of some minor dimensions and of the exact shape of the shell and webs is left to the student.

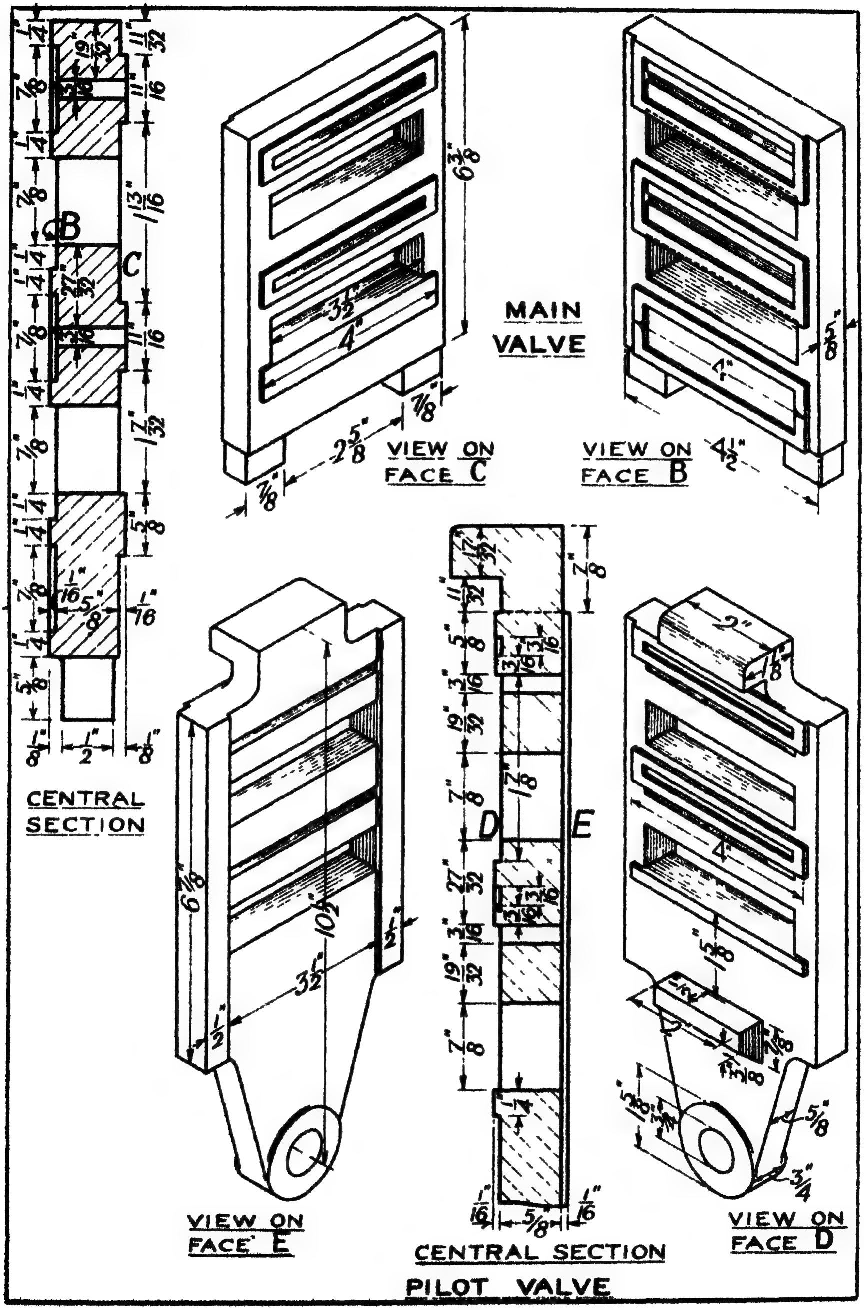


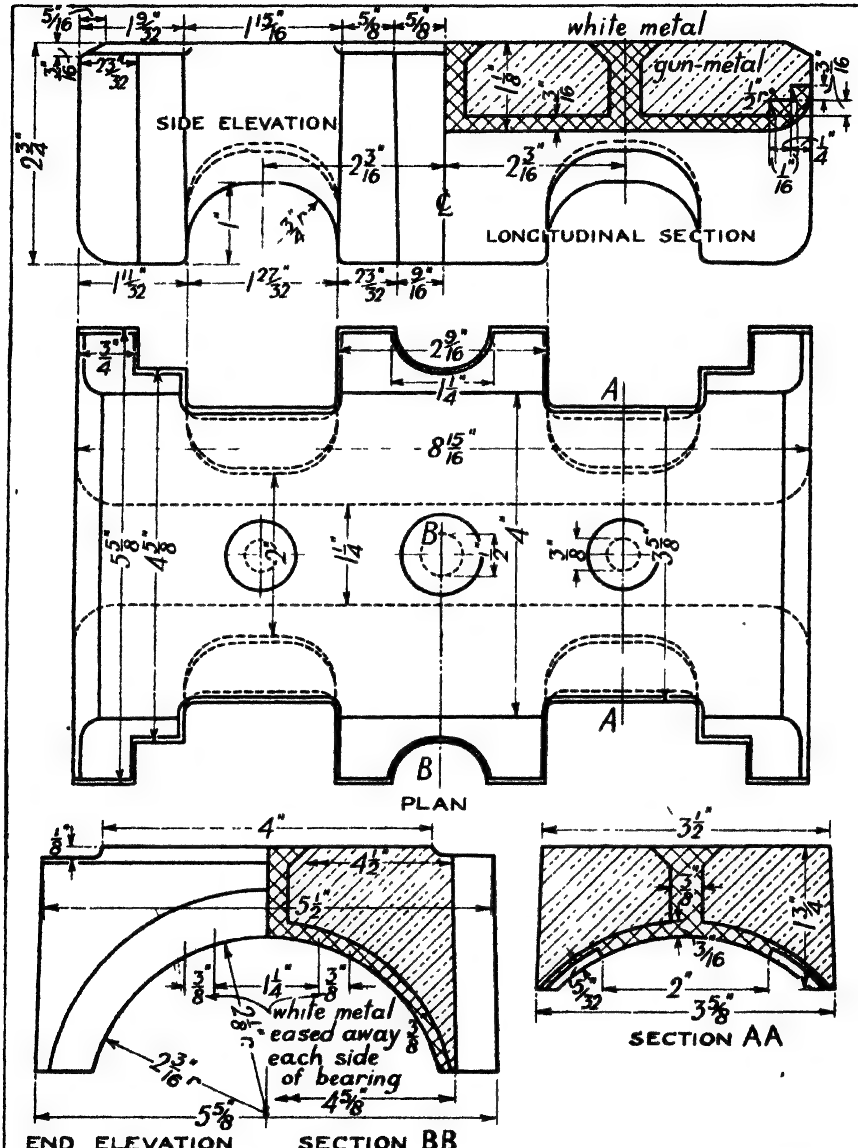

ONE HALF OF VALVE BODY, WITH GUIDE

When assembled, face B of the main valve slides on A, and face D of the pilot valve on face C of the main valve, the latter lying between the projections on the pilot valve. The two valves slide between guides. Draw, half size, the following undimensioned views of both valves in position on the valve body and fully open to steam: sectional elevation; view on E; and plan, half in section through a port. [$3\frac{1}{2}$ hr.]

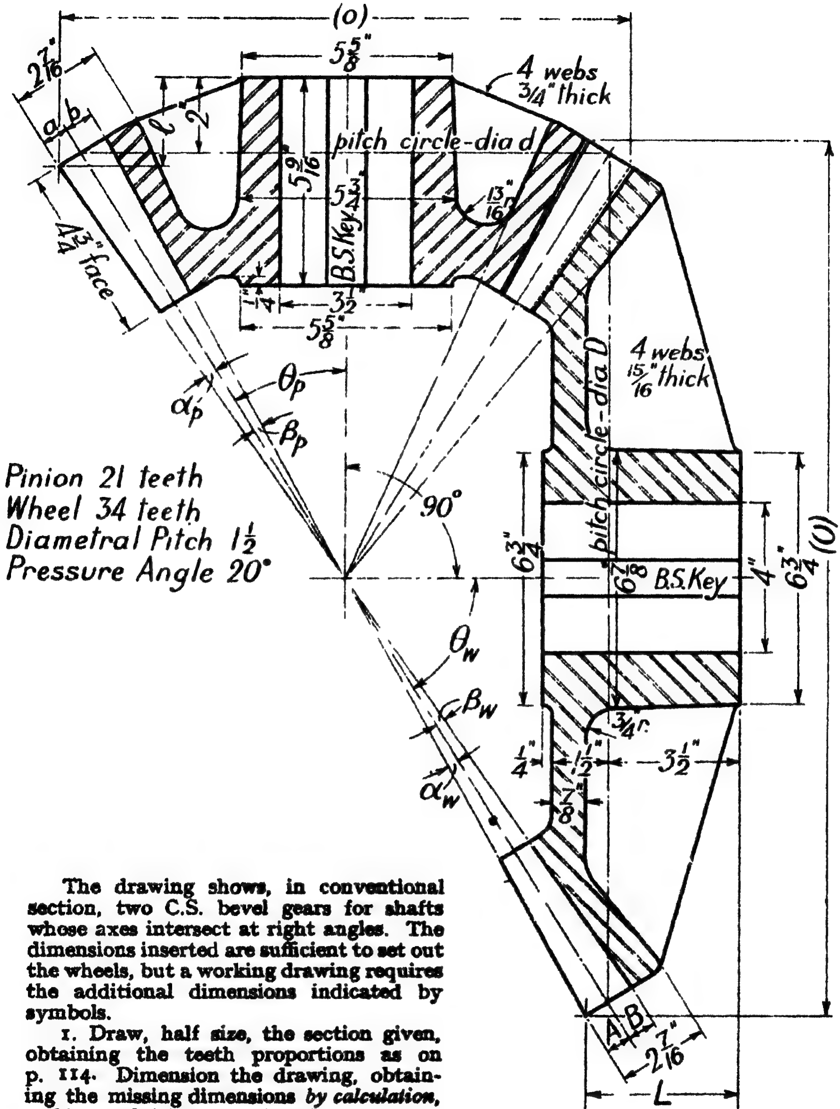
Draw also, full size, sections of the valves and body on cardboard strips and examine the action. Show that the total travel of the pilot valve = $1\frac{1}{8}$".

Materials: body and main valve, semi-steel; pilot valve, Phos.B.; guides, W.I.; bolts, steel. (Refer to p. 130.)


MAIN
VALVE
VIEW ON
FACE C
VIEW ON
FACE B
CENTRAL
SECTION
VIEW ON
FACE E
CENTRAL SECTION
VIEW ON
FACE D
PILOT VALVE




Working drawings are given above of an axle box bearing for a 12-ton wagon, L.M.S. Rly. Co. (refer to p. 104). Draw, full size, the following dimensioned views: elevation, half in section; half plan and half inverted plan; end elevation; sectional end elevation, half on A A and half on B B. [3½ hr.]

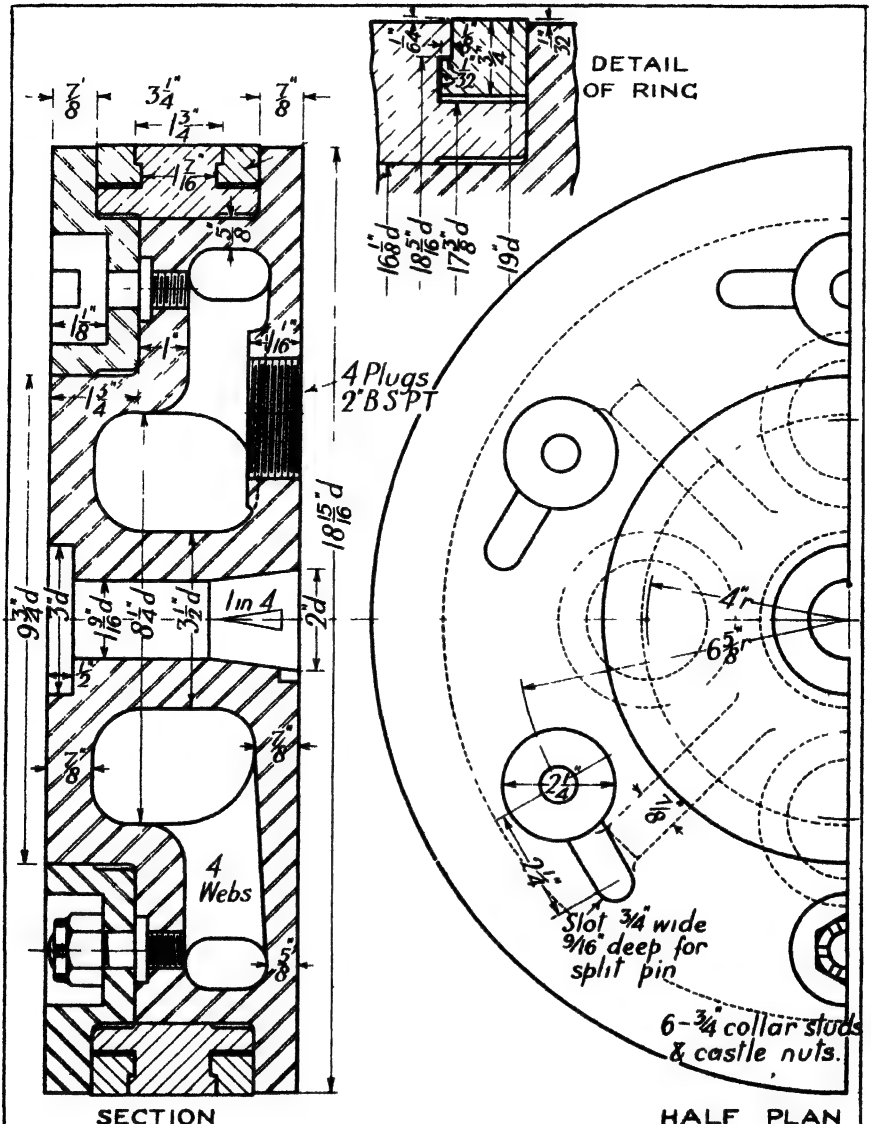



The drawing shows, in conventional section, two C.S. bevel gears for shafts whose axes intersect at right angles. The dimensions inserted are sufficient to set out the wheels, but a working drawing requires the additional dimensions indicated by symbols.

1. Draw, half size, the section given, obtaining the teeth proportions as on p. 114. Dimension the drawing, obtaining the missing dimensions *by calculation*, and insert finish marks. [3 hr.]

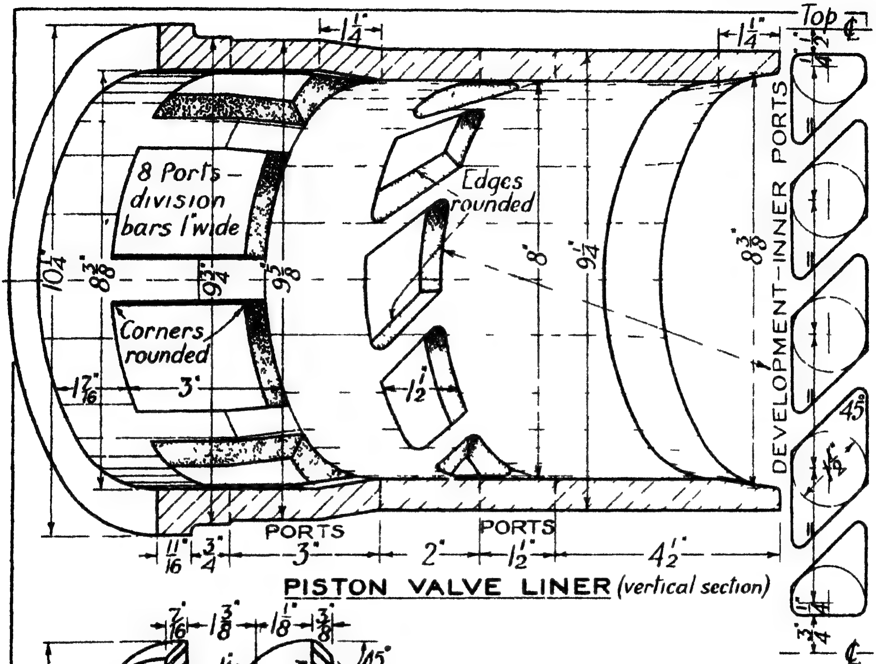
2. Draw, half size, an elevation of the pinion, inverted, and a half plan, on the teeth. [3 hr.]

Answer to (1): D = 22·66″; O = 23·348″; $\theta_w$ = 58° 18′; L = $4\frac{1}{16}$″;
$d$ = 14·00″; $o$ = 15·125″; $\theta_p$ = 31° 42′; $l$ = $2\frac{11}{16}$″;
A = $a$ = ·666″; B = $b$ = ·771″; $\alpha_w = \alpha_p$ = 2° 52′; $\beta_w = \beta_p$ = 3° 18′.

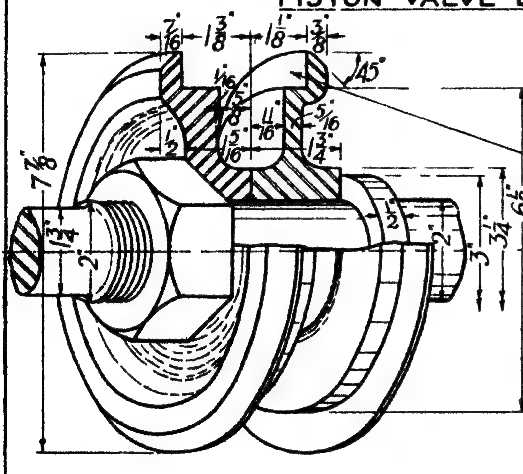



**Draw, half size, the following dimensioned views of the assembled piston: side elevation, half in section through a web; end view on the junk ring, half in section to reveal the cavity. Use your own judgment for parts not dimensioned. [2½ hr.]**

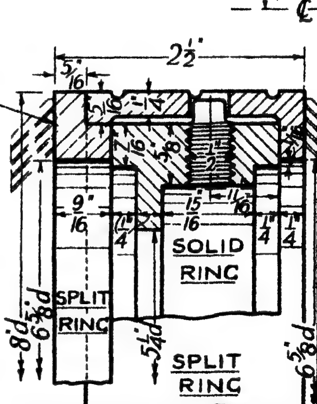
**Note.—The sectional view given is conventional and is not strictly accurate.**




PISTON VALVE LINER (vertical section)



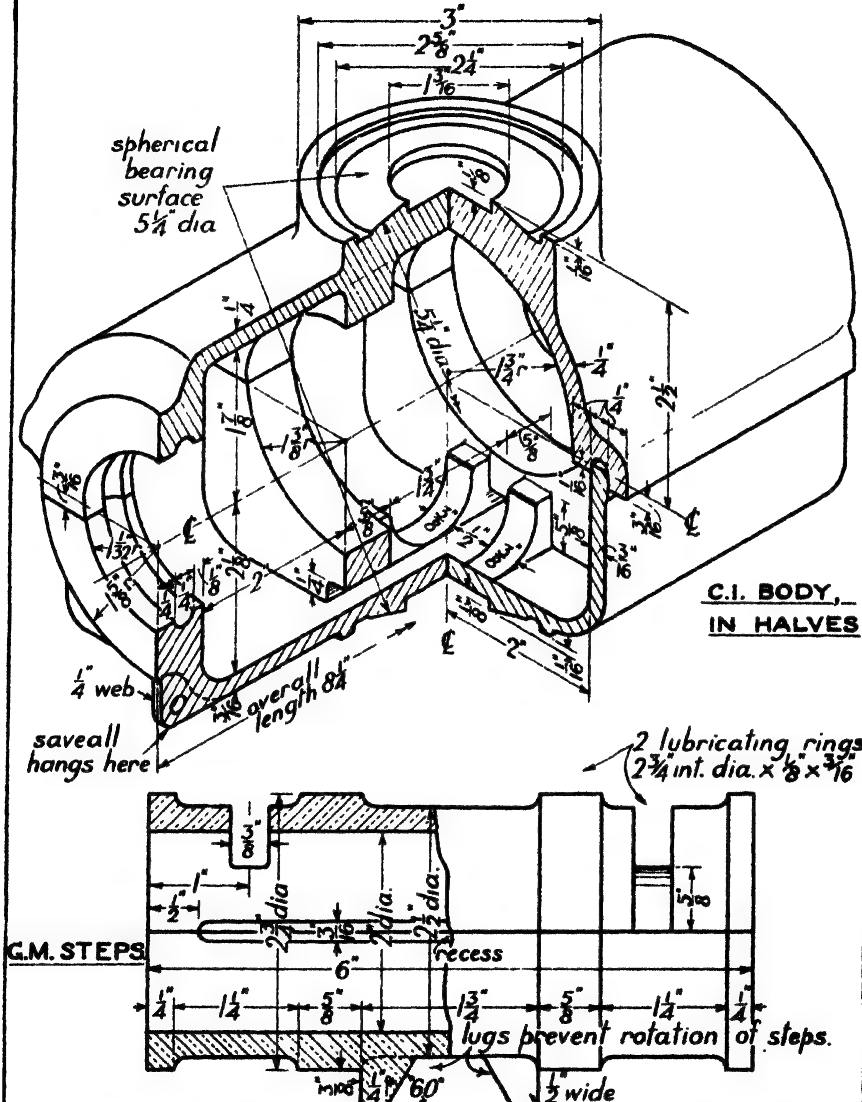
PISTON VALVE HEAD



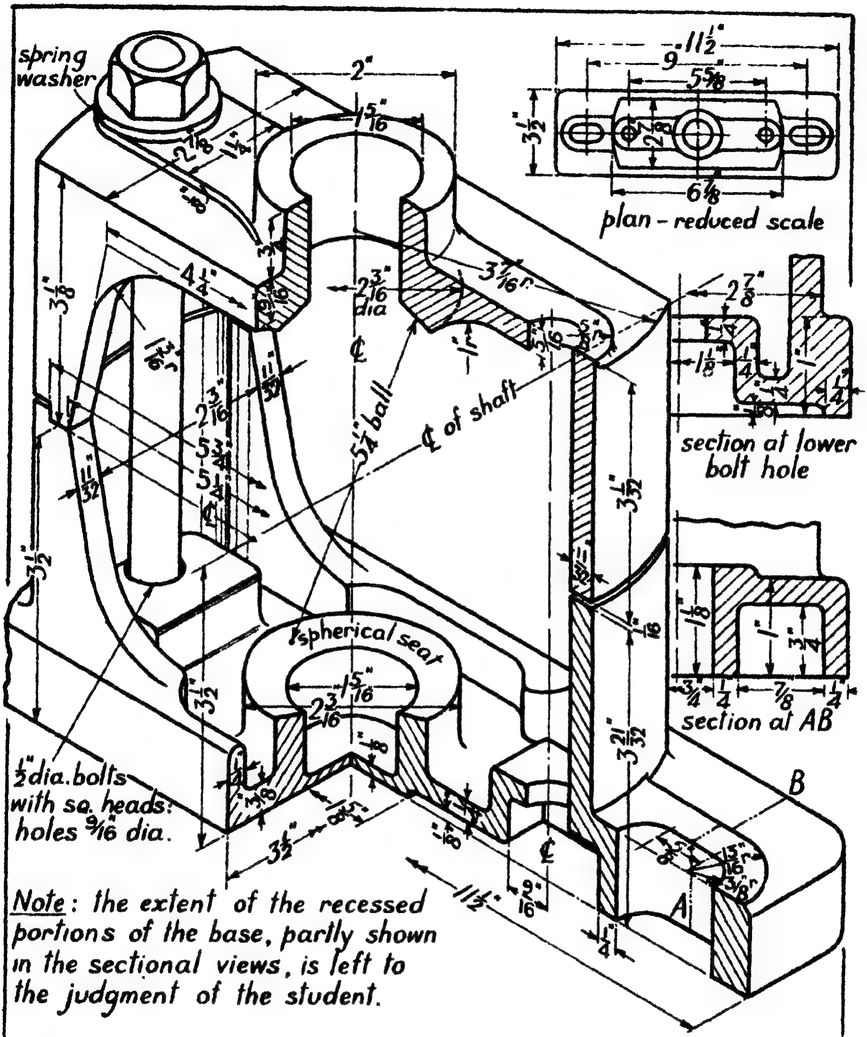

SECTION OF RINGS AT TOP

**Refer to the cylinder on p. 147. Three C.I. rings, one solid and two split, fit between the halves of the valve head. The smaller split ring and the solid ring are located by set screws (not shown) in the valve head at the bottom; a set screw in the top of the solid ring locates the larger split ring. The valve head is keyed to the spindle and secured by nut and cotter.**

**Draw, full size, the following dimensioned views of the valve head arranged within the liner, between the ports: elevation, liner and upper half of valve in section; half end view from left to right, in section through the larger ports. [3½ hr.]**




*Note :- the following have been omitted to simplify the drawing : filling hole and cap, drain cock, oil level indicator, oil savealls.*




**Particulars are given of a self-oiling swivel bearing and its pedestal. Prepare the following drawings, full size, dimensioning (1) and (2) but not (3).**

**(1) Swivel Bearing: longitudinal sectional elevation; sectional end view through centre. [2 hr.]**

**(2) Pedestal: side elevation, half in section; end elevation, half in section; plan on lower half. [3 hr.]**

**(3) Assembled Bearing and Pedestal: side elevation; end elevation; plan. [2½ hr.]**

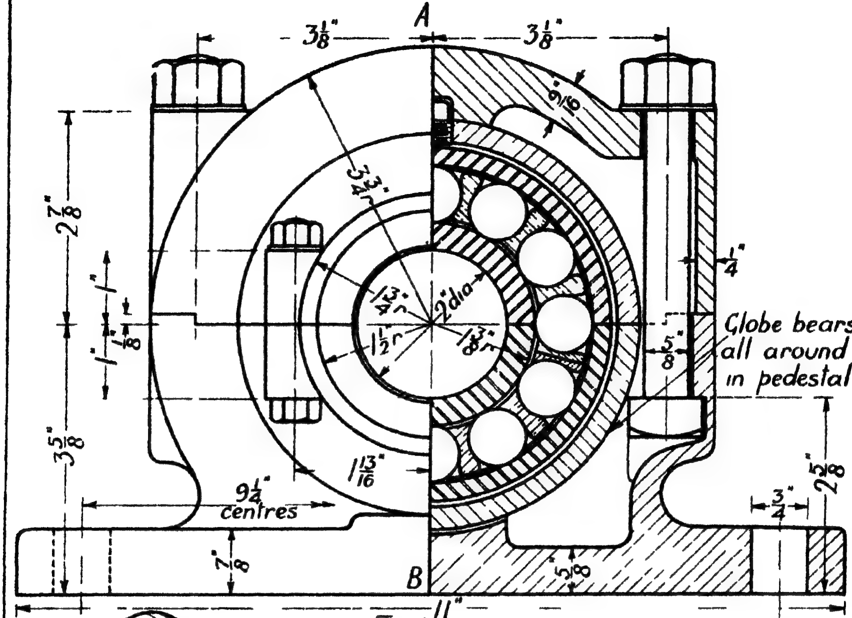



Fig 1

ELEVATION OF BEARING

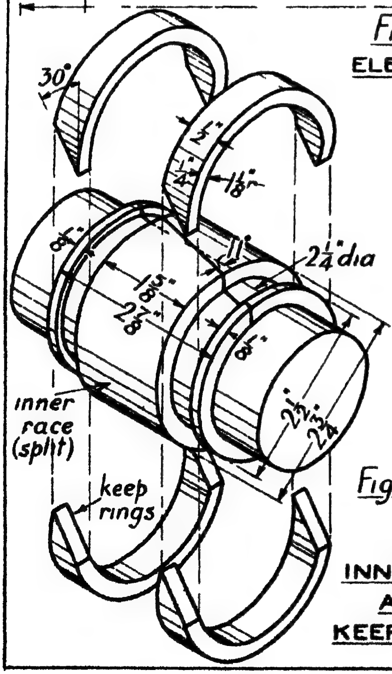



Fig 2

INNER RACE AND KEEP RINGS

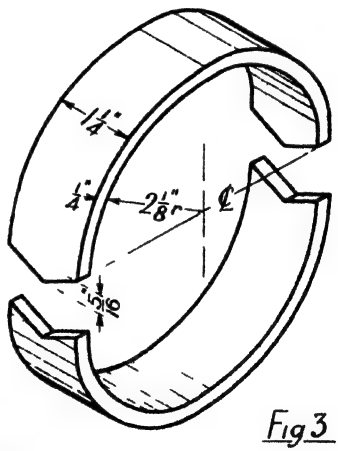



Fig 3

OUTER RACE

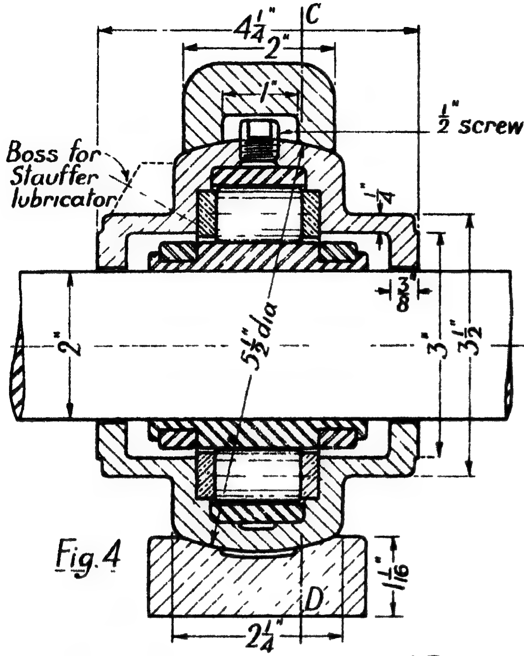



Fig. 4

SECTIONAL ELEVATION ON AB

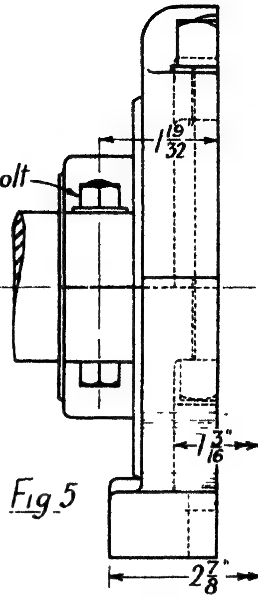



Fig. 5

HALF END ELEVATION

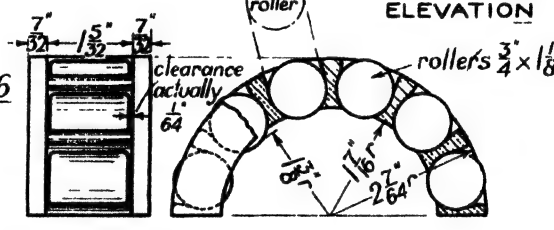


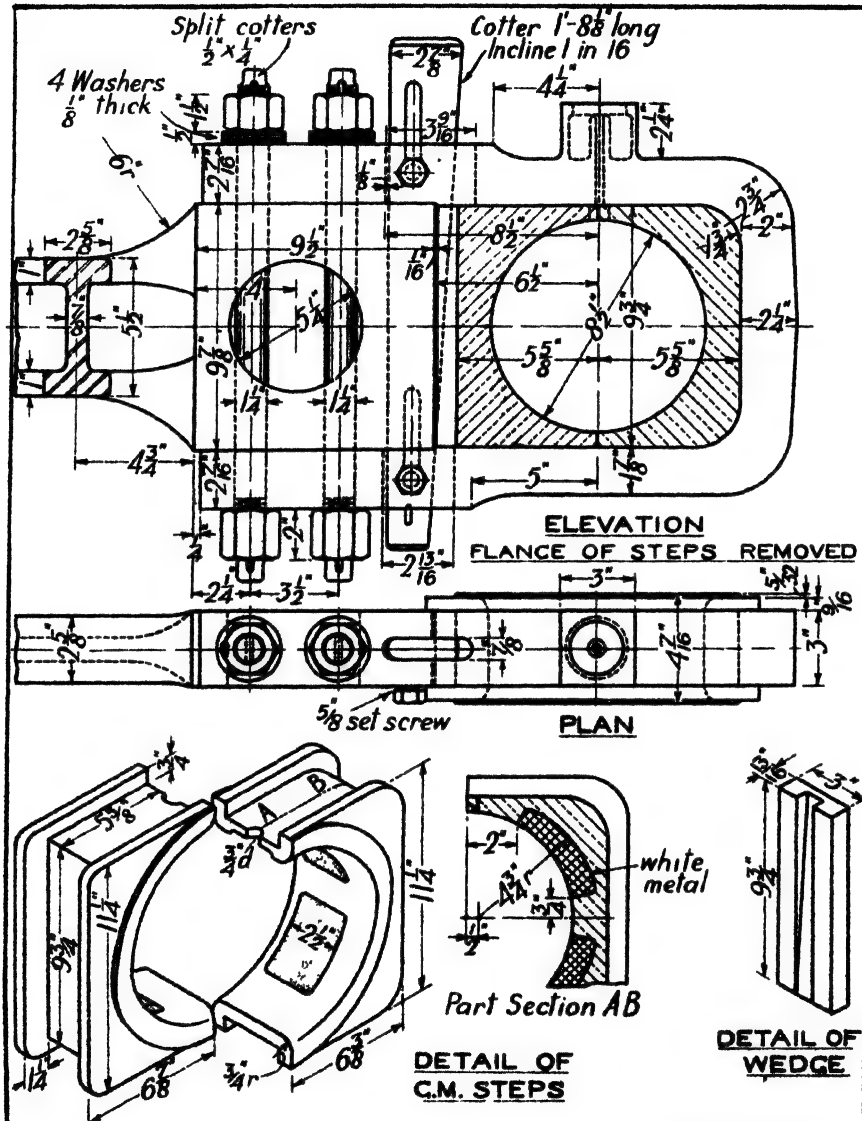

Fig. 6

DETAIL OF CAGE

The drawings give particulars of a Cooper split roller bearing for a 2-inch diameter shaft. The halves of the steel split inner race (fig. 2) are held together by two steel keep rings. These rings are split diagonally so that the distance between the horns is slightly less than the diameter of the sleeve: they are driven over the sleeve, which they grip firmly. The rollers are held in two semicircular brass cages, one of which is shown in fig. 6. The steel split outer race (fig. 3) is a press fit in the halves of the C.I. housing. The housing is turned spherically and is free to swivel between the halves of the pedestal.

Draw, full size, the following views of the assembled bearing: elevation, as in fig. 1, but with one half in section on *CD*; end elevation, half in section on shaft axis; plan. Dimension the views. [5 hr.]

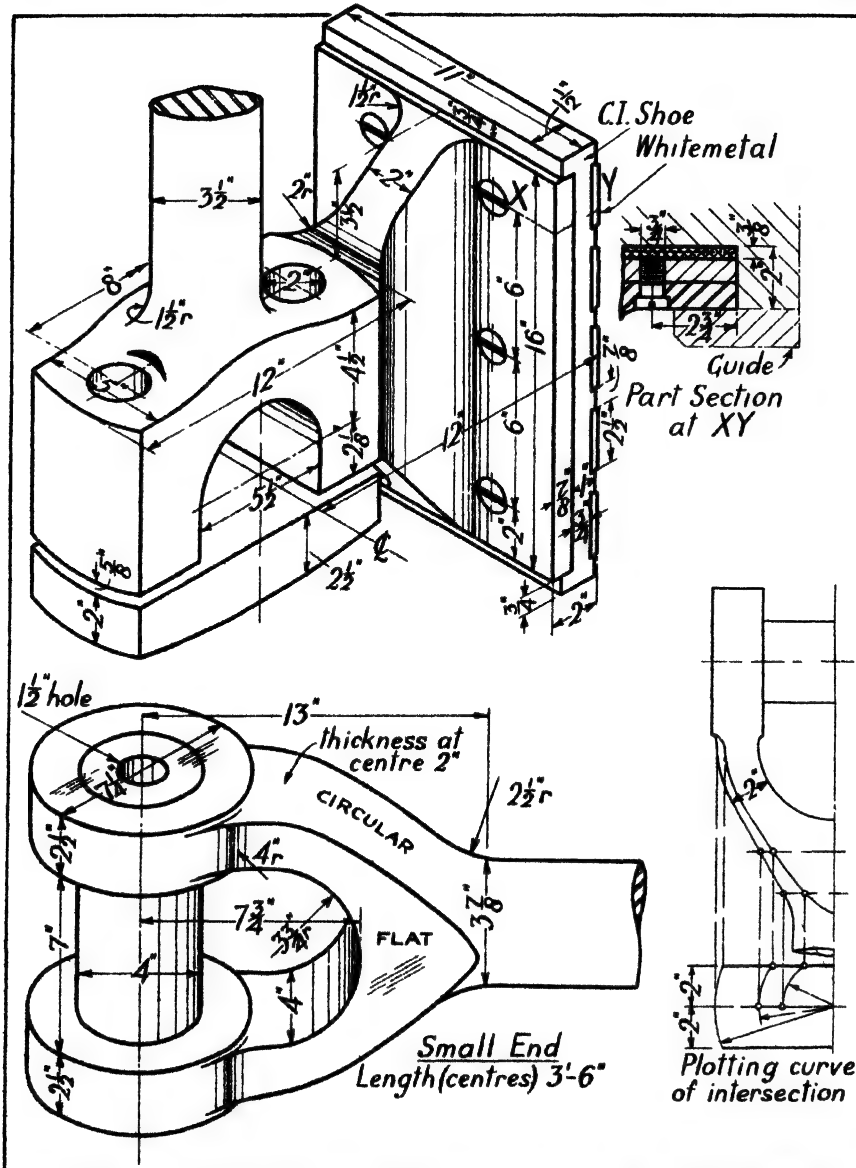
Note.—Many minor dimensions have been omitted intentionally: the student should use his own judgment in settling these.




Details are given above of the big end of the locomotive connecting rod shown on p. 143.

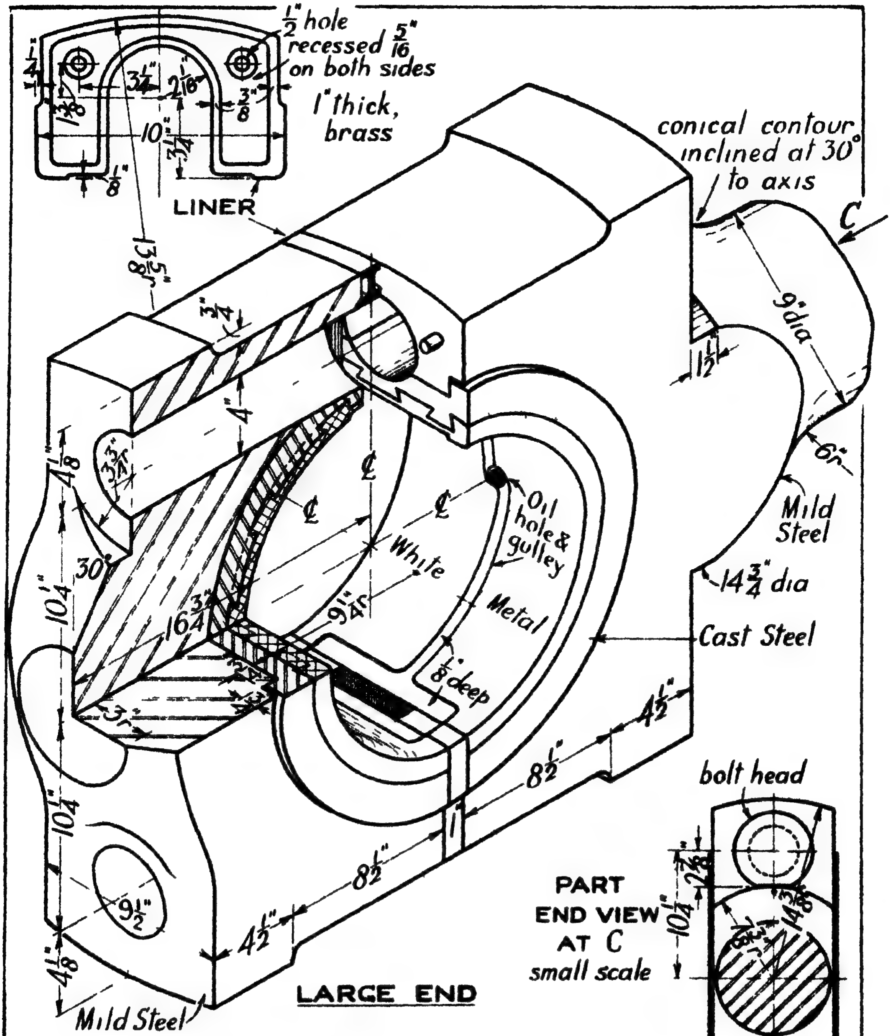
(1) Draw, half size, the following views: elevation, with upper half in section to show the cotter; sectional plan on the rod axis; end view from right to left. Insert the principal dimensions. [3 hr.]

(2) Prepare completely dimensioned working drawings of the steps. [1½ hr.]




(1) Draw, quarter size, an elevation, end view, and plan of the crosshead with the connecting-rod end in position. Show suitable G.M. bearings and devise a locking arrangement for the bolts. Insert dimensions. [3 hr.]

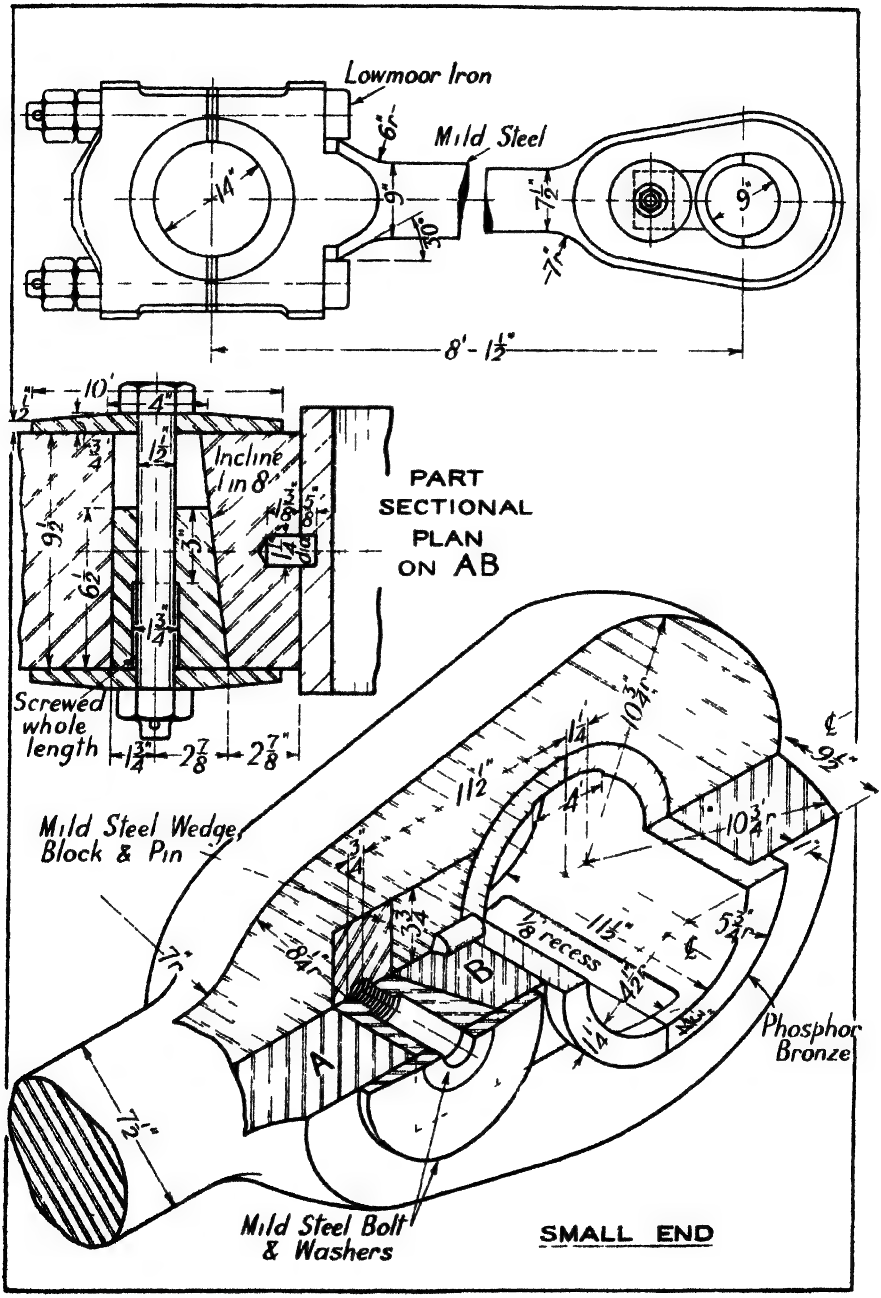
(2) Prepare working drawings of the complete connecting rod, the lower end of which is 4½" dia. and similar to that shown on p. 141. [4 hr.]

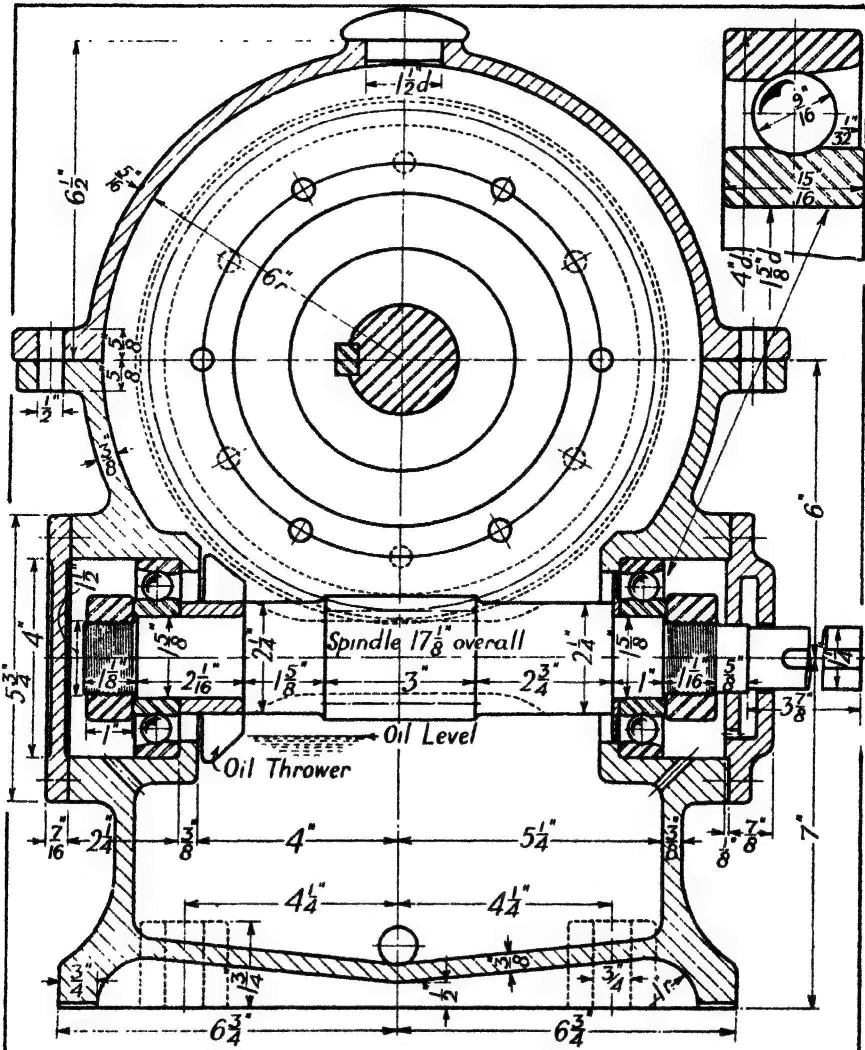



Details of the bearing for the large end are given on p. 105, and the bolts are shown on p. 53 Lubrication fittings have been omitted for simplification. Prepare the following:

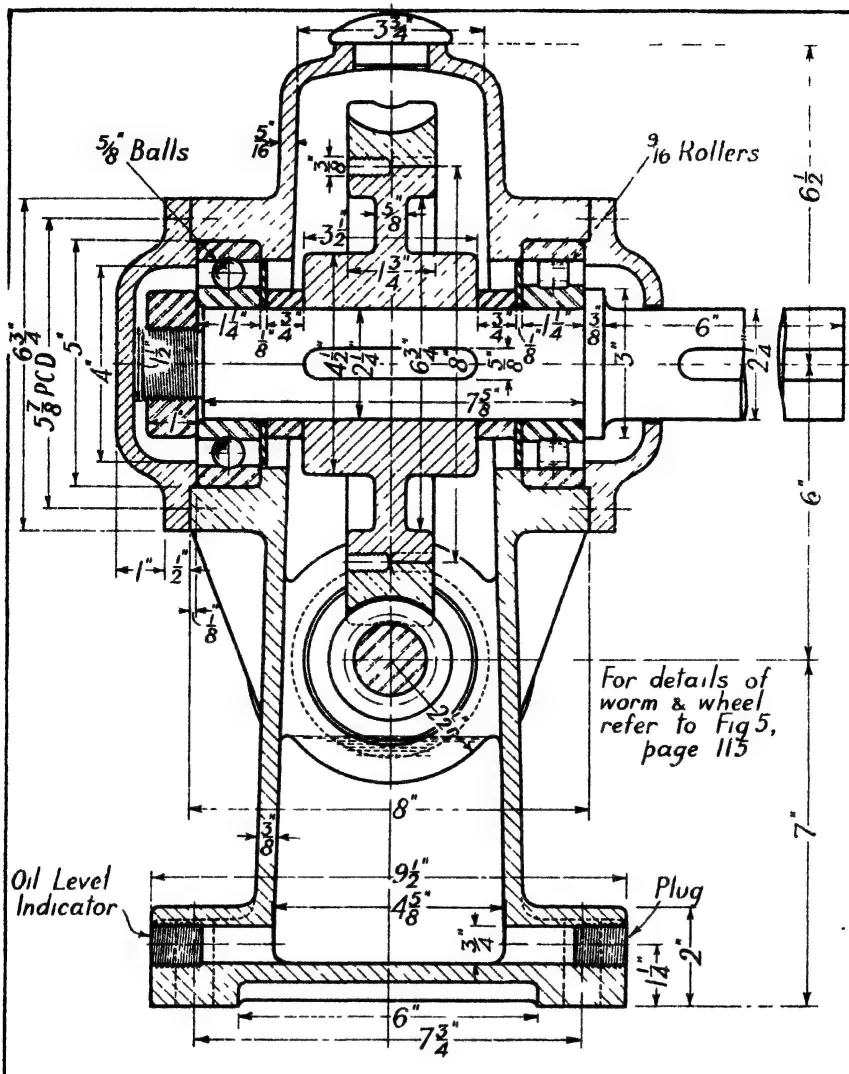
(1) Separate drawings of each end of the rod complete with bearings, showing, in each, an elevation with the upper half in section, a plan, and a view on the outer end. Use third angle projection, Scale, quarter size. Insert principal dimensions. [Large end 2 hr.; small end 1½ hr.]

(2) A working drawing of the finished connecting rod—ex cap, bearings, and fittings. Use third angle projection. Scale, eighth size. [2½ hr.]


Lowmoor Iron
Mild Steel
14"
6"r
9"
30°
7½"
7"r
9"
8'-1½"
10"
4"
½"
¾"
1½"
Incline 1 in 8
PART SECTIONAL PLAN ON AB
9½"
6½"
3"
1¾"
1¼" dia
Screwed whole length
1¾"
2⅞
2⅞"
Mild Steel Wedge, Block & Pin
Mild Steel Bolt & Washers
Phosphor Bronze
10¾"r
1¼
11½"
4"
9½"
¾"
3¾"
8¼"r
7"r
⅛" recess
11½"
5¾"r
4½"
1¼"
7½"
A
B
SMALL END

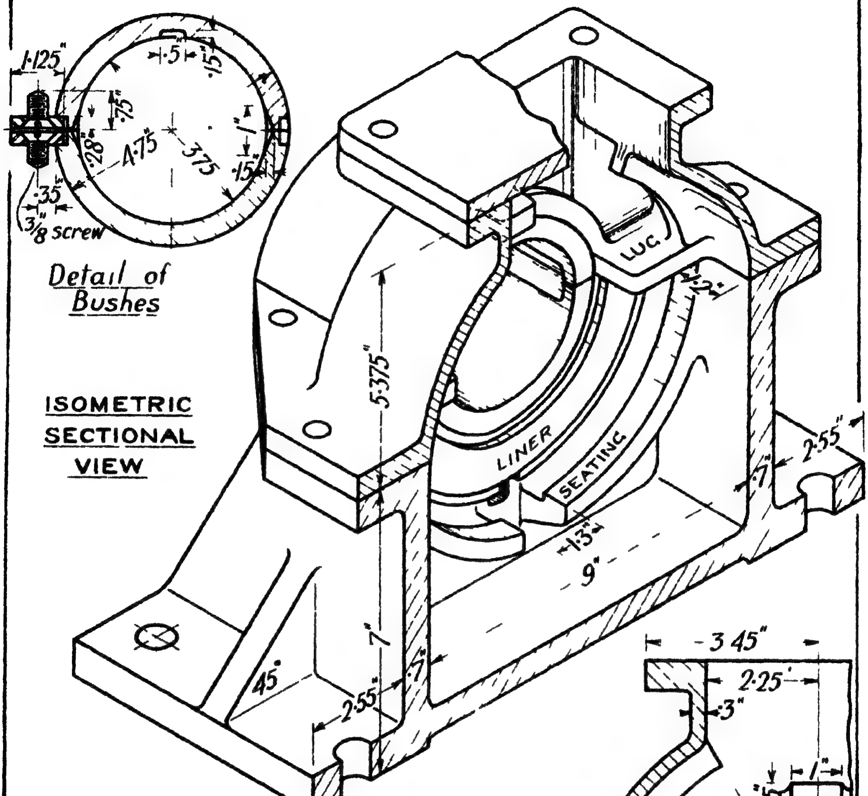



The drawings above and opposite show a worm reduction gear complete with C.I. casing. The lower part of the casing forms a reservoir for oil, the level of which is kept below the worm shaft to prevent leakage. An oil thrower revolving with the worm supplies lubricant to the wheel teeth. The wheel is of C.I., with a G.M. rim shrunk on and secured by 12 pins screwed half in the rim and half in the wheel. The casing is symmetrical.



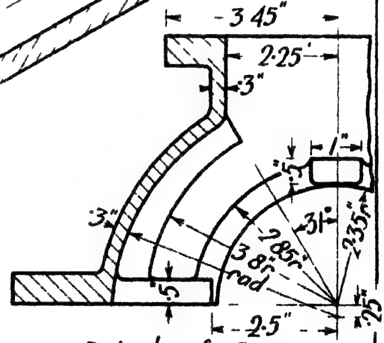

Prepare, half size, the following drawings: (*a*) views corresponding to those given but with one half of each showing an *outside* view; (*b*) a half plan with the cover, wheel, and shaft removed. Dimension the casing only. [7–8 hr.]

Note.—Many details and dimensions are left for the student to settle by way of exercise; e.g. numbers, positions, and diameters of casing joint bolts; locking devices for shaft nuts; &c.




Detail of Bushes

ISOMETRIC SECTIONAL VIEW



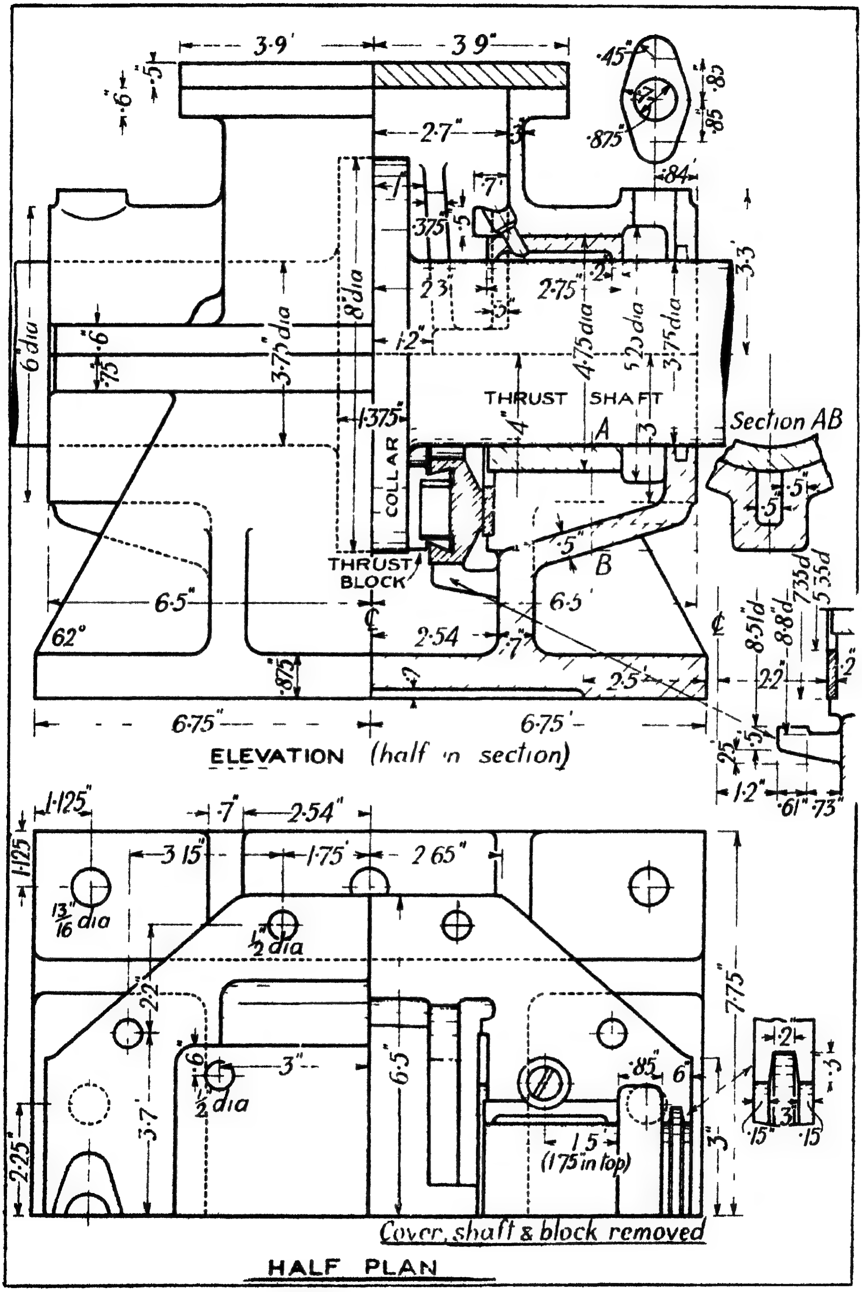

Detail of Cover

The drawings give details of a symmetrical C.I. housing for the Michell thrust bearing shown on p 111. The shaft takes a thrust in each direction and requires a thrust bearing on both sides of the collar. Each bearing is supported on a semicircular seating, and presses on a steel liner. Lugs cast on the cover prevent the bearings and liners from turning with the shaft.

(1) Draw, half size, the following views, and dimension them: complete sectional elevation, omitting the thrust blocks and shaft; end view, half in section at a central plane; half plan. [5 hr]

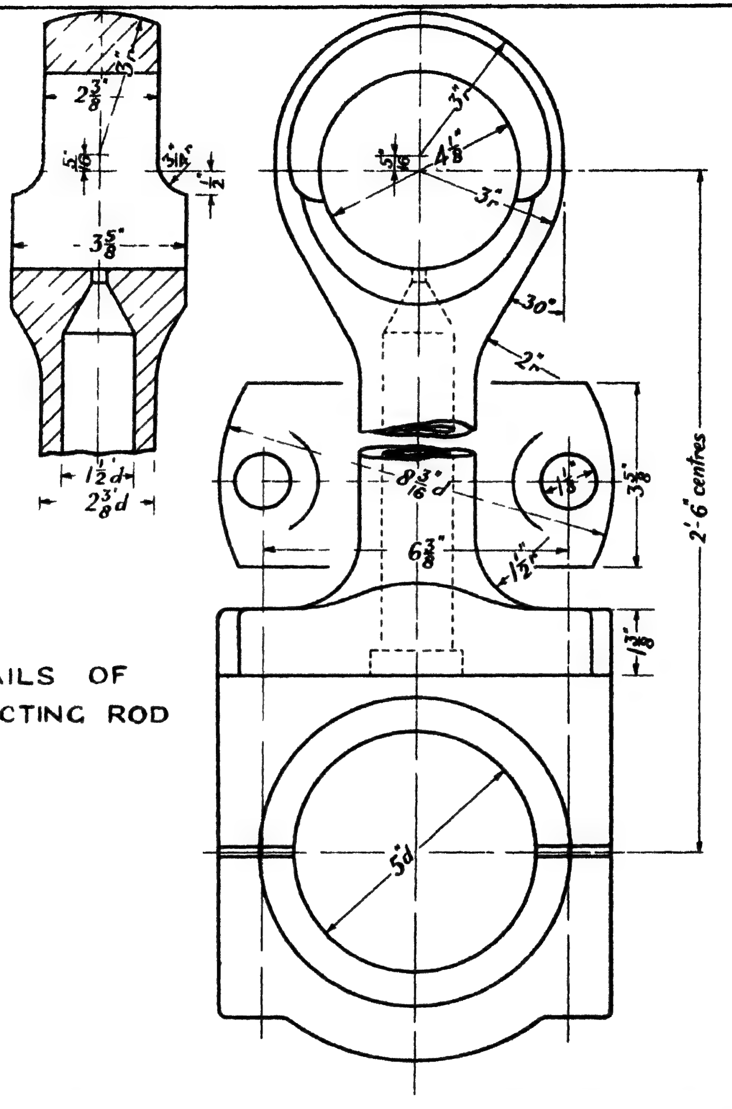
(2) Prepare a working drawing of the cover, giving, with other views, one from the underside. [3½ hr.]

Note.—The actual design has been simplified somewhat in the drawings given. For example, the following have been omitted. drain plug, air vent, oil-level gauge, oil catcher, and oil scraper; the last two are supported on the ribs and lugs cast on the cover.




ELEVATION (half in section)

HALF PLAN

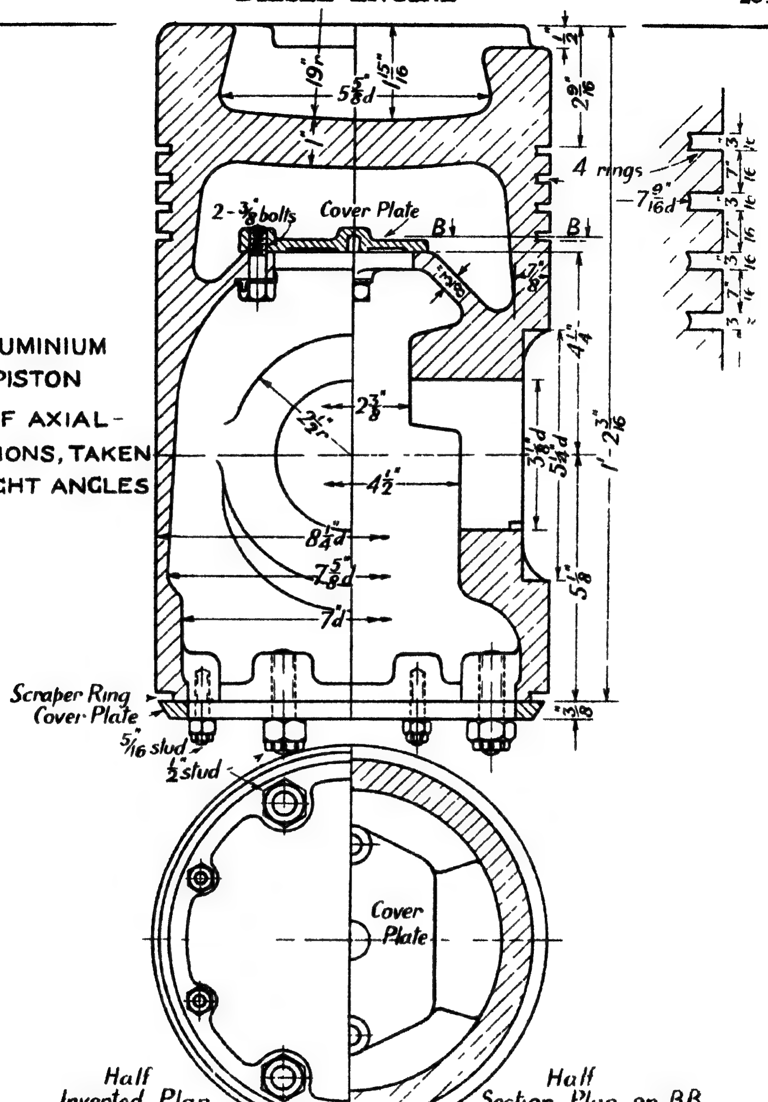



DETAILS OF CONNECTING ROD

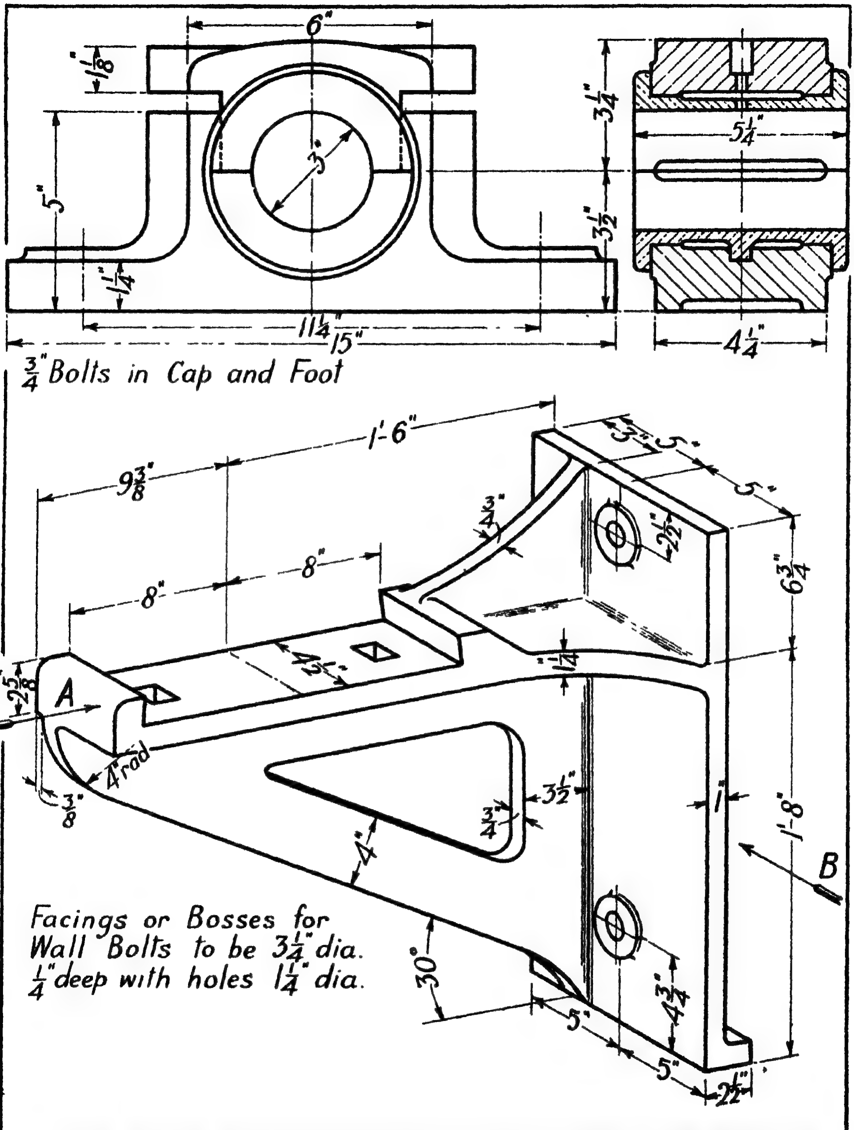
The drawings show the general design and principal dimensions of an aluminium alloy piston and steel connecting rod for a 12" stroke airless injection oil engine. The small-end bush, not shown, is of G.M. shrunk into the eye, with the outer half reduced in width. The large-end bearing is of C.S. lined with W.M. and is shown only in outline.

ALUMINIUM PISTON

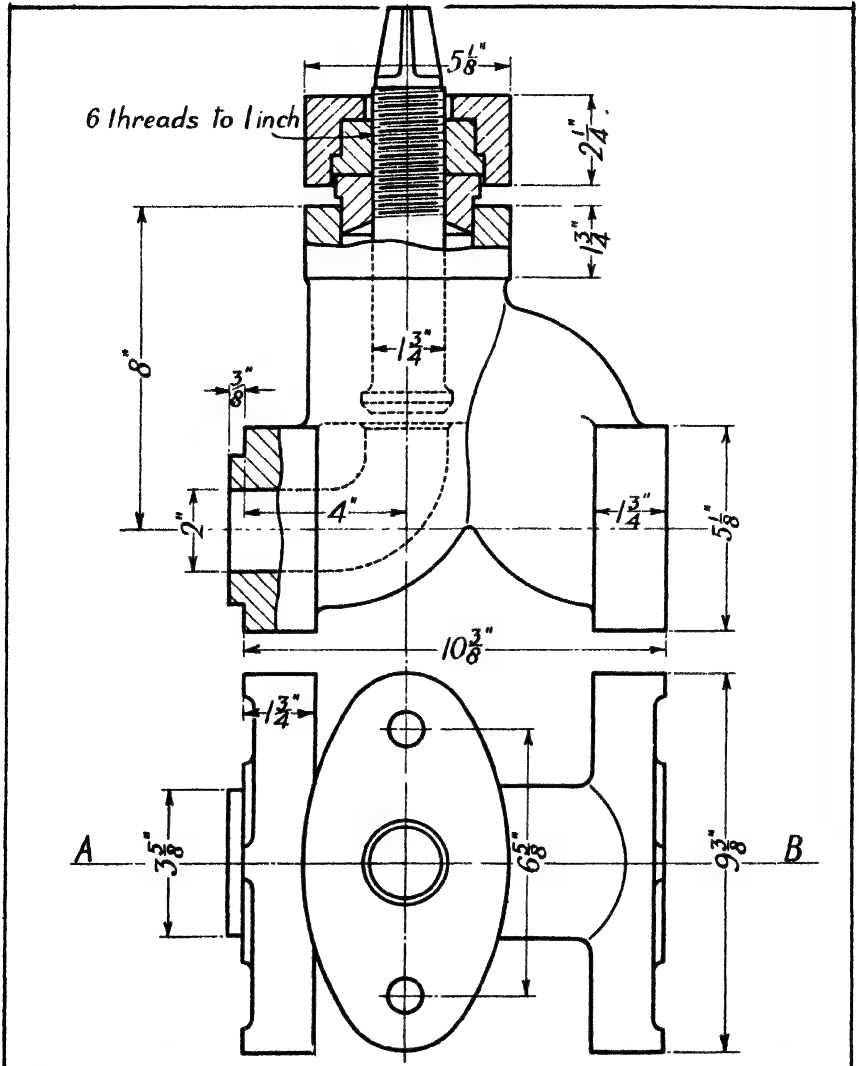
HALF AXIAL-SECTIONS, TAKEN AT RIGHT ANGLES




Prepare completely dimensioned drawings of both piston and connecting rod, using your own discretion for proportions undimensioned. The gudgeon pin, $3\frac{1}{8}''$ dia., is to be prevented from turning, and shaped cover-plates $5\frac{1}{4}''$ dia are to be bolted to the piston to give oil-tight joints at the pin ends. Complete both big- and small-end bearings for the connecting rod, lightening the former in the manner shown on p. 141. [10 hr.]

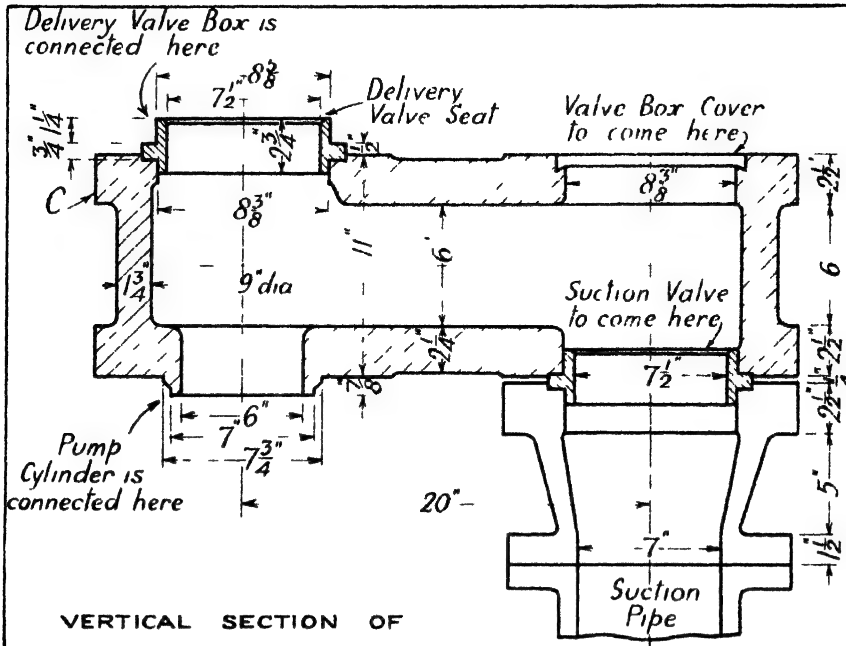



Draw, half size, an elevation of the **wall bracket** looking in the direction of the arrow B. Also draw the **pedestal bearing** in position on the bracket. The pedestal is to be in section, the cutting plane to contain the axes of the holding-down bolts. Bolts and nuts should be shown. Crosshatch the sections of the parts and use your judgment in estimating dimensions which are not given. Also on the right of the above view, draw an end elevation of the bracket (without the pedestal) looking in the direction of the arrow A. Line in this view and add the leading dimensions. [Time, approx. 3 hr.] *Wh. Sc. Exam.*




An elevation and plan of a **hydraulic stop valve** are given above. The thickness of the metal in the shell or body of the valve is $\frac{3}{4}$", the diameter of each gland bolt is $\frac{7}{8}$", and the axial length of the threaded part of the valve spindle is $3\frac{1}{2}$".

Draw, half size, a complete section of the valve cut by a vertical plane containing the line AB. Crosshatch neatly, freehand, the sections of the parts, and use your judgment in estimating dimensions which are not given. Show how the valve seat is fixed in position and draw the threads in section. Draw to the right of the above view an elevation of the valve, looking from left to right. Line in this view and dimension it. [Time, approx. 3 hr.] *Wh. Sc. Exam.*




VERTICAL SECTION OF
SUCTION VALVE BOX AND CONNECTOR
TO PUMP CYLINDER AND DELIVERY VALVE BOX

The following sketches of parts of a **high-pressure pump** are give above and opposite:

(1) A vertical section, plan and end elevation of the delivery valve box.

(2) A section of the valve-box cover.

(3) A section of the packing ring.

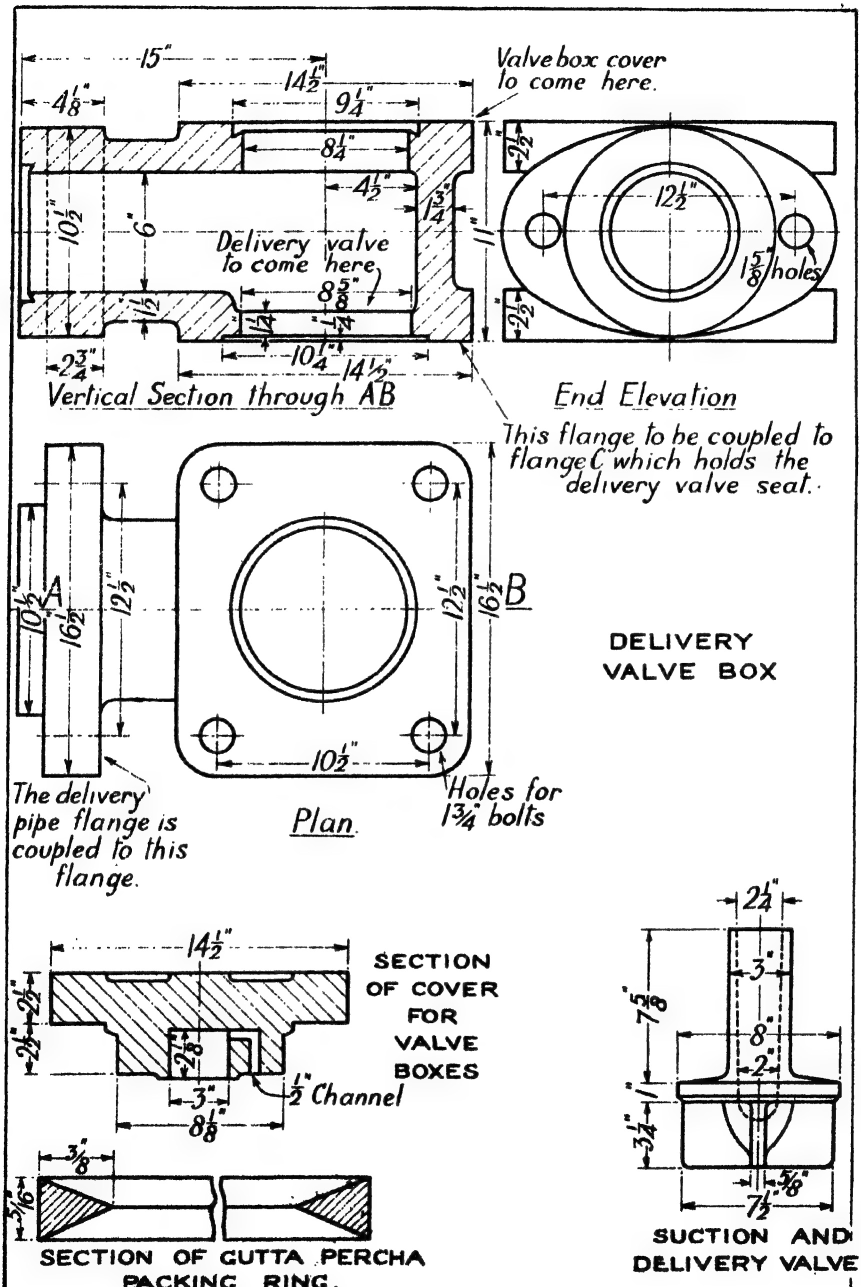
(4) An elevation of a valve; and

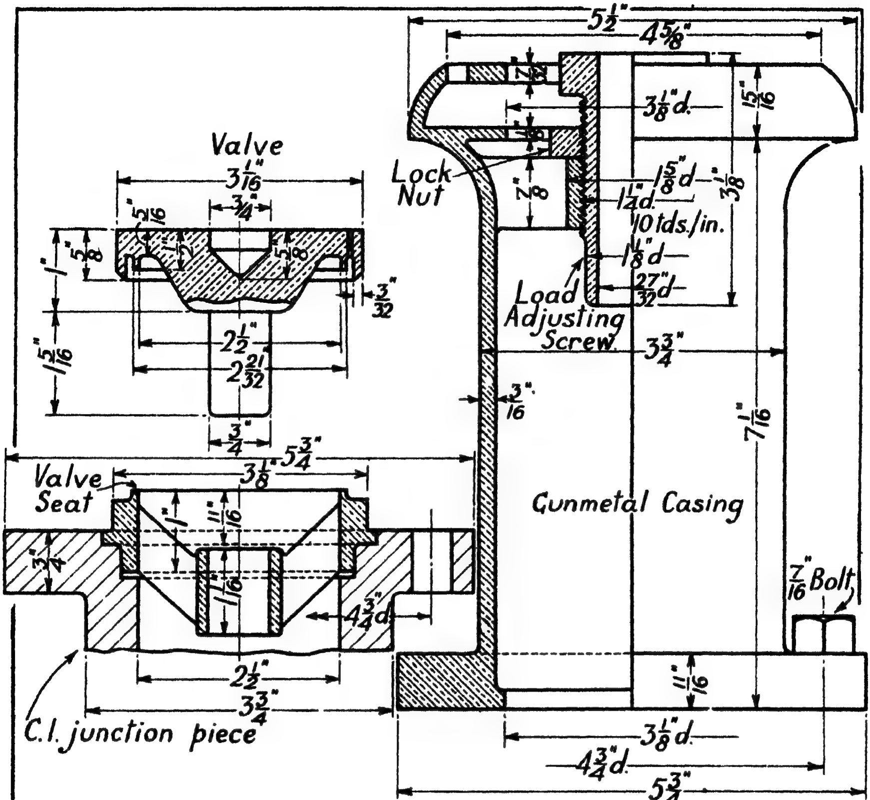
(5) A vertical section of the combined suction valve box and the connector to the pump barrel and delivery valve box.

Make the following drawings to a scale of quarter full size:

(*a*) Reproduce in section the suction valve box with the valve, cover, and suction-pipe flange assembled. Add to this drawing an elevation of the connector and the delivery valve box and cover. Dotted lines and nuts may be omitted, but centre lines should be shown.

(*b*) Below (*a*) draw a plan of the parts already on your paper. Line in and add a few dimensions to this view. [Time, approx. 3 hr.] *Wh. Sc. Exam.*

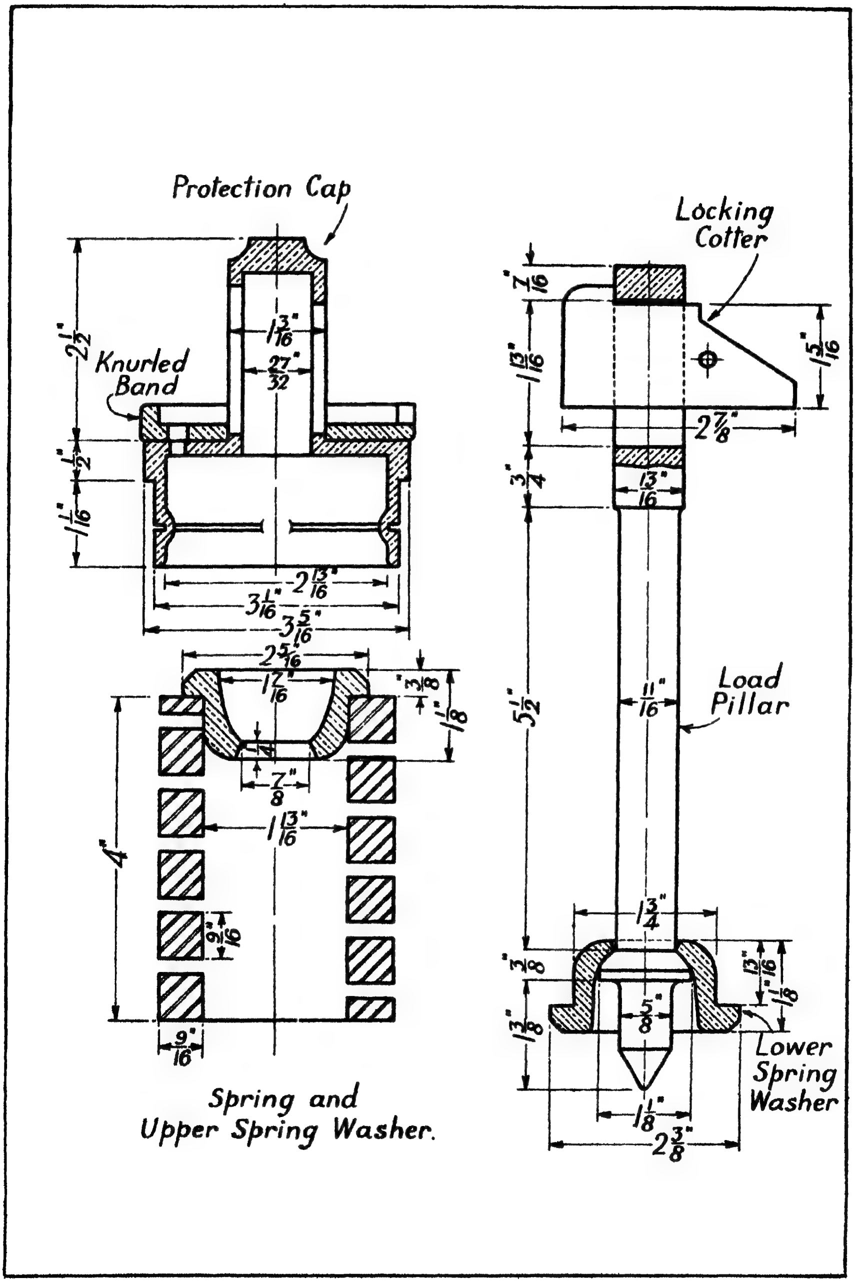

Valve box cover to come here.
Delivery valve to come here
Vertical Section through AB
End Elevation
1 5/8" holes
This flange to be coupled to flange C which holds the delivery valve seat.
A
B
The delivery pipe flange is coupled to this flange.
Plan.
Holes for 1 3/4" bolts
DELIVERY VALVE BOX
SECTION OF COVER FOR VALVE BOXES
1/2" Channel
SECTION OF GUTTA PERCHA PACKING RING.
SUCTION AND DELIVERY VALVE



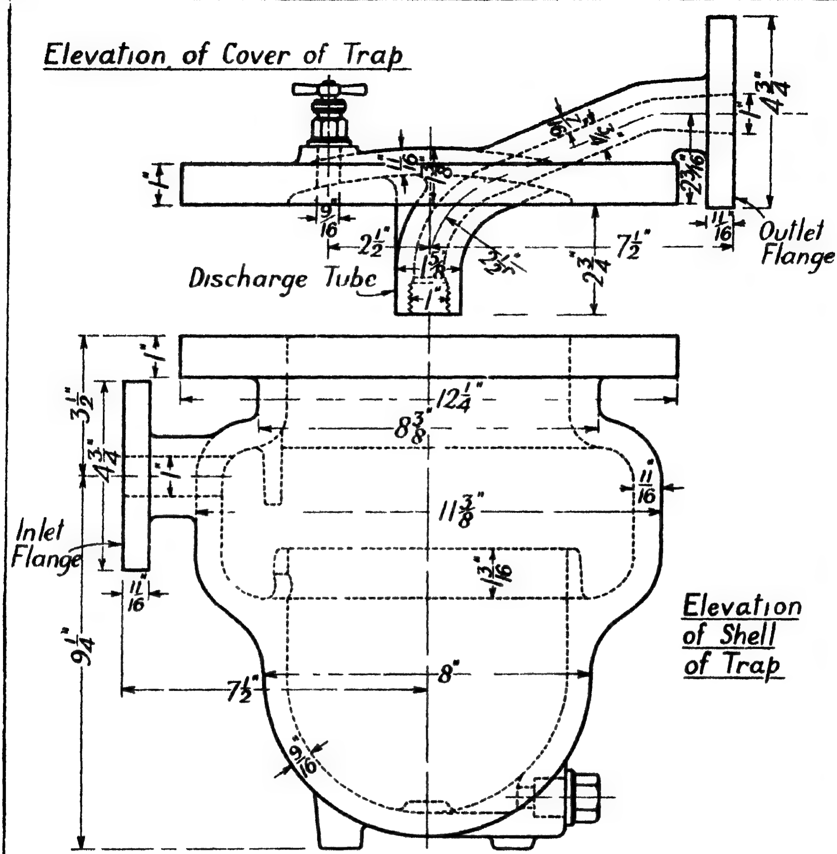

Dimensioned sketches of the detailed parts of a **"pop" safety valve for a steam boiler** are given. Draw the parts assembled ready for attachment to a boiler, giving a vertical axial section of the valve, valve seat, upper spring washer, and outer shell or casing; but showing the spring, lower spring washer, the load pillar, the adjusting screw, and the protecting cap in elevation.

You may omit the cast-iron junction piece, and the curved outlines of the spring may be replaced by straight lines in the customary manner. The spring section is incorrectly drawn in the figure.

Line in the valve seat, the spring and the load pillar, and dimension them. Explain how this safety valve differs in its action from that of one fitted with a mushroom valve. [Time, approx. 3 hr.] *Wh. Sc. Exam.*




Spring and Upper Spring Washer.

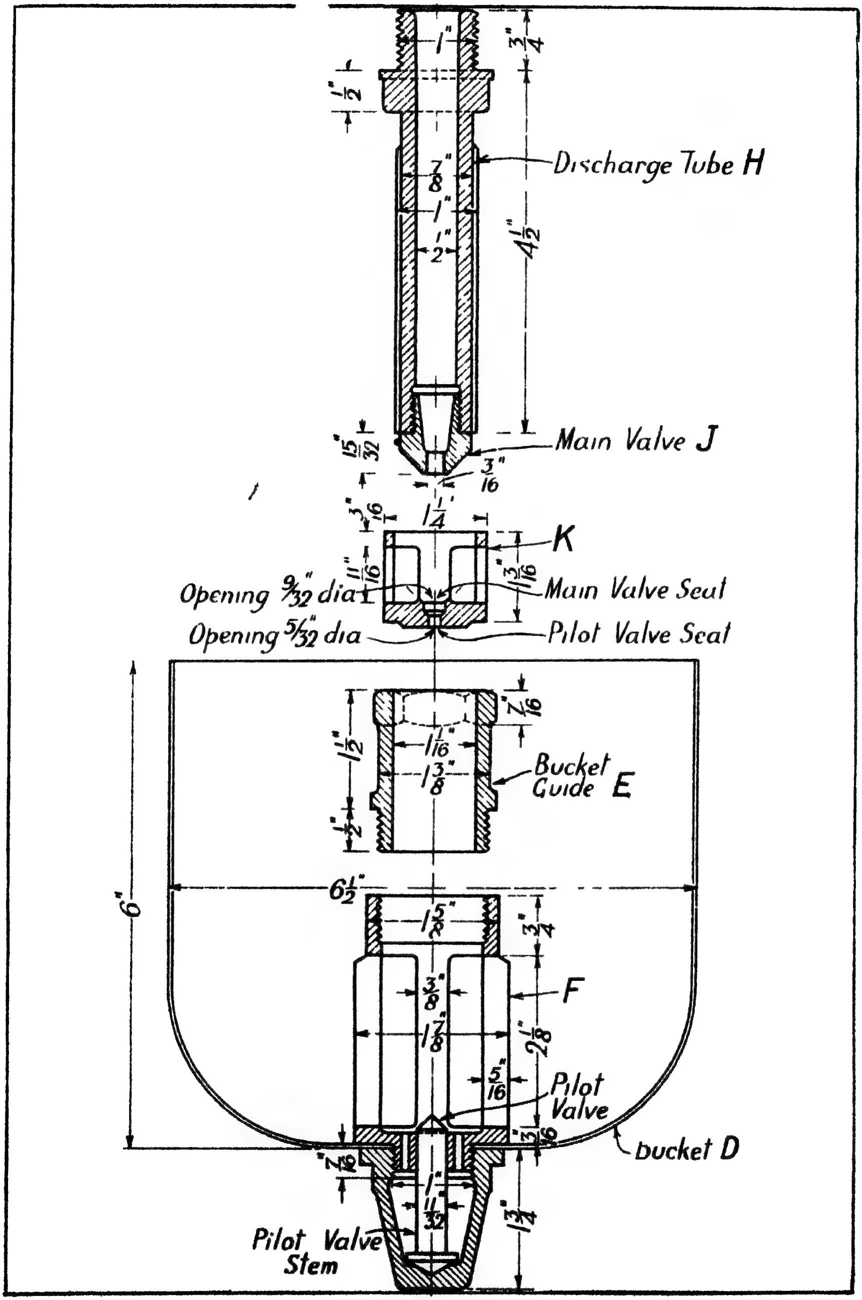


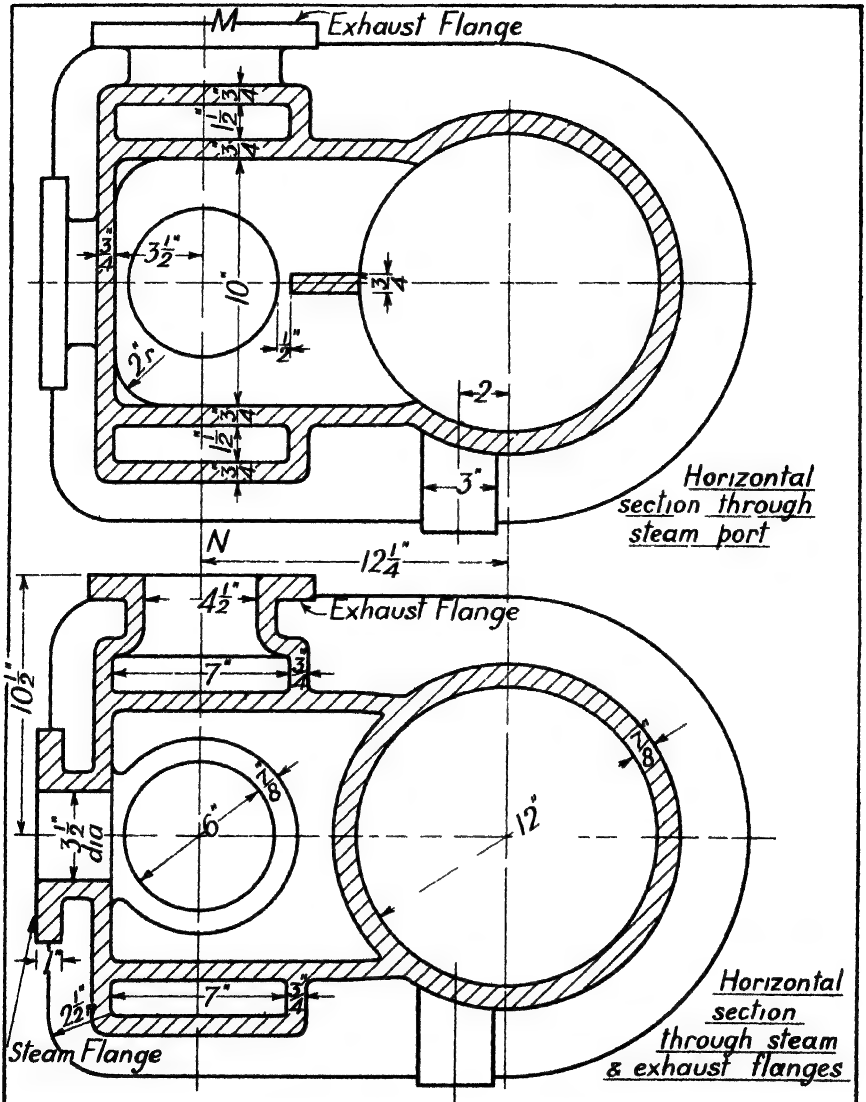

Some dimensioned sketches of the details of a **bucket steam trap** are given.

(*a*) Draw, half size, an axial section of the shell of the trap with the cover in position, the cutting plane to be parallel with the paper.

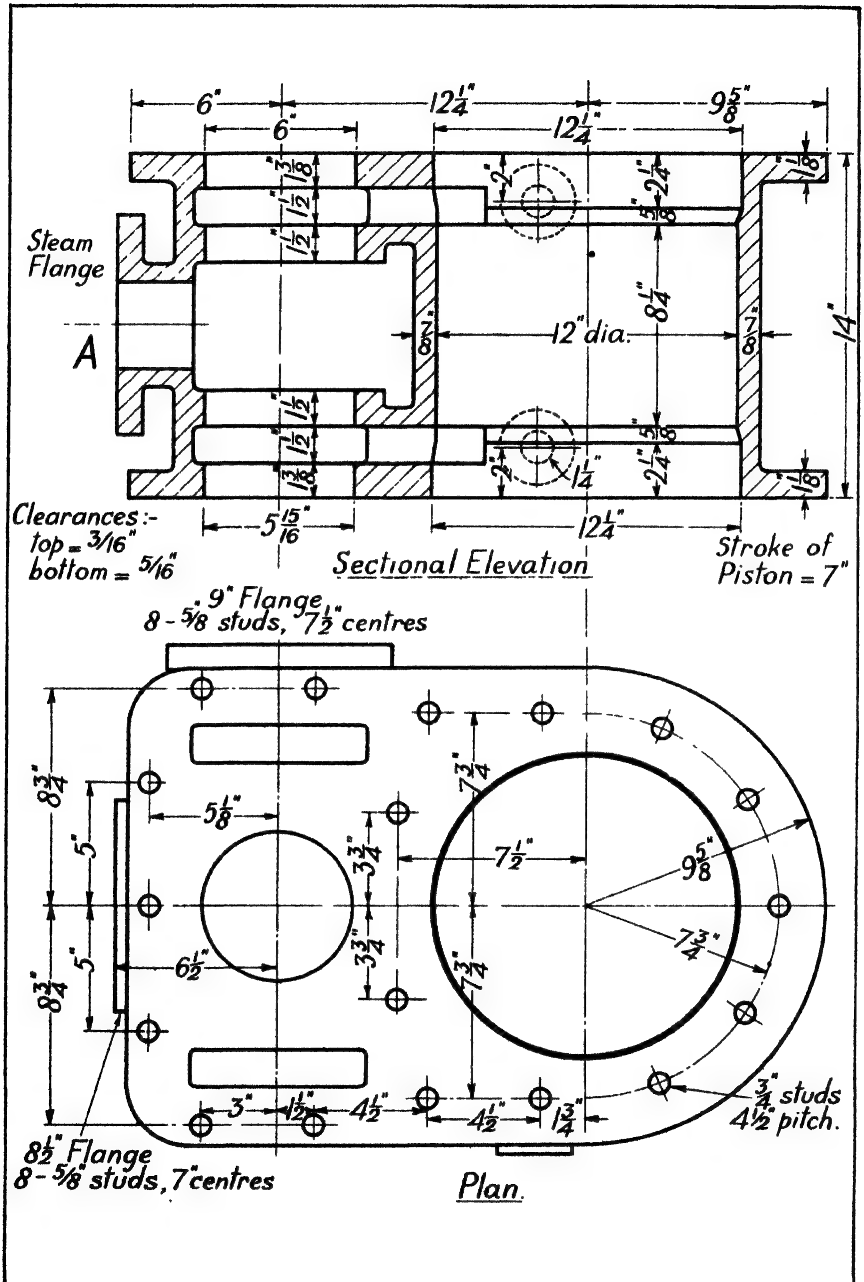
(*b*) Also draw, full size and assembled ready for operation, an axial section of the bucket D, the bucket guide E and F, the pilot valve and valve seat K, the main valve J, and the discharge tube H. This drawing is to be made entirely separate from (*a*).

The diameter of the pilot valve opening is $\frac{1}{16}''$ and the weight of the sliding parts may be taken as $3\frac{1}{2}$ lb. To simplify calculation, the bucket guide may be taken as having a uniform outer diameter of $1\frac{5}{8}''$. The steam pressure is 150 lb./in.$^2$ by gauge At what distance from the top edge of the bucket is the surface of the water inside of it when the bucket begins to move downwards? Explain in a very few words how the trap works. [Time, approx. 3 hr.] *Wh. Sc. Exam.*


Discharge Tube H
Main Valve J
K
Opening 9/32" dia
Main Valve Seat
Opening 5/32" dia
Pilot Valve Seat
Bucket Guide E
F
Pilot Valve
Bucket D
Pilot Valve Stem




Draw, half size, the vertical section A complete with piston, cylinder covers, valve chest covers, piston valve, and liners. Line in the piston and piston valve in your drawing and add the necessary dimensions to those details. Also sketch freehand a vertical section of the valve chest casting, the cutting plane containing the line MN. The piston valve may be omitted from this view. [Time, approx. 2½ hr.] *Wh. Sc. Exam.*


Steam Flange
A
Clearances:-
top = 3/16"
bottom = 5/16"
Sectional Elevation
Stroke of Piston = 7"
12" dia.
14"
9" Flange
8 - 5/8 studs, 7½" centres
8½" Flange
8 - 5/8" studs, 7" centres
3/4" studs
4½" pitch.
Plan.

**(1)** Fig. 1 gives a sketch of a **hydraulic cylinder and piston,** the gauge pressure on the top of the piston being 1000 lb./in.$^2$, while the atmosphere has free access to the bottom of the piston through the cylinder open end. Design the cylinder, flange, and studs, the cylinder being of gunmetal, for which a safe working tensile stress is 3500 lb./in.$^2$. The studs are of steel, with a safe working stress of 7000 lb./in.$^2$ at the bottom of the thread. For the cylinder thickness, use the formula

$$\text{thickness} = \frac{d}{2}\left(\sqrt{\frac{f+p}{f-p}} - 1\right),$$

where $d$ = internal dia., $f$ = stress, and $p$ = internal pressure.

*Glas. Univ., B.Sc.*

**(2)** A combined **pump and piston rod** for a direct-acting pumping engine is shown in fig. 2. The rod is subjected to a load of 32 tons, alternately tensile and compressive, and is in two parts connected by a sleeve and cotters. Design the rod and cottered joint for the following conditions:

Allowable tensile stress in screwed ends and across cotter holes, 4 tons/in.$^2$.

Allowable shear stress in the cotter, 3 tons/in.$^2$.

Allowable crushing stress on cotters and rod shoulders, 7 tons/in.$^2$.

The diameter of the rod must be such that, when tested by Rankine's formula, the buckling load is not less than 70 tons.

$$\text{Buckling load P (tons)} = \frac{21\text{A}}{1 + \dfrac{l^2}{7500k^2}}.$$

(The stuffing-box packings through which the rod passes are of the freely floating type, and therefore have no effect upon the resistance of the rod to buckling.)

*Glas. Univ., B.Sc.*

**(3)** An unsymmetrical **balance weight,** in the form of a cast-iron block, is attached to a crank web by two long bolts, fig. 3. The bolts are in clear holes, and therefore incapable of carrying any shear force, any tendency of the block to move in the plane of the crank web being resisted by the projecting flanges. Calculate the load on the bolts when the shaft is running at 300 revs./min. and choose a suitable diameter from the following table:

| | | | | | | |
|---|---|---|---|---|---|---|
| Dia. of bolts, in | $\frac{5}{8}$ | $\frac{3}{4}$ | $\frac{7}{8}$ | 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ |
| Safe load, lb. | 570 | 1060 | 1710 | 2630 | 3660 | 5100 |

Weight of cast iron, 0·28 lb /in.$^3$.

*Glas. Univ., B.Sc.*

**(4)** The end plate of a boiler supported from the shell plate by a **gusset stay** is shown in fig. 4. The area of the end plate supported is 540 in.$^2$; the boiler pressure is 200 lb./in.$^2$. The riveting of the angles to the shell must not reduce the strength of the plate by more than 20 per cent.

Design the gusset stay for the following stresses:

Tensile and shear for rivets, 9,000 lb./in.$^2$.

Tensile for gusset plate, 11,000 lb./in.$^2$.

Shear for angles, - - 10,000 lb./in.$^2$.

Bearing for rivets - - 9 tons/in.$^2$.

Allowance for double shear to be taken as $1\frac{7}{8}$ times that for single shear.

*Glas. Univ., B.Sc.*

**Note.**—For the above hand sketches only were required, approximately to scale, and properly dimensioned. Time for each question about 35 min.

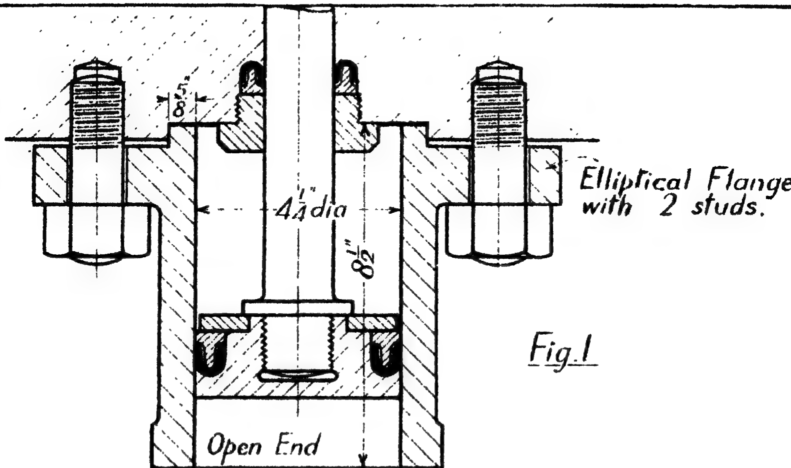



Fig. 1

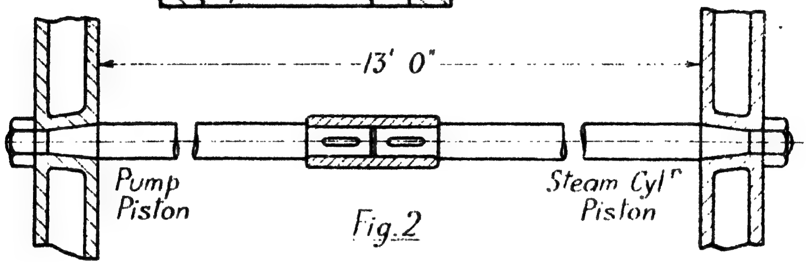



Fig. 2

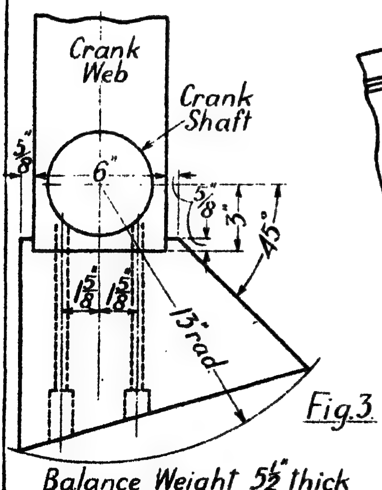



Fig. 3

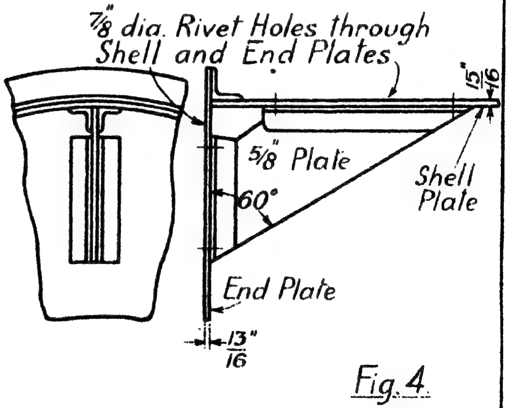



Fig. 4

**(1) A horizontal hydraulic cylinder** is shown in fig. 1, the piston of which gives motion to two side rods through a mild-steel crosshead, cottered to the piston. The "inside" end of the piston is fitted with a single U leather packing, and there is an annular space of $\frac{1}{2}''$ between the piston and cylinder walls. The power for the forward stroke is obtained by admitting water at 750 lb./in.$^2$ pressure at the end A; on the return stroke, A is opened to exhaust, the full pressure of 750 lb./in.$^2$ being maintained in the annular space during both strokes.

Determine the load for each stroke, and design and give fully dimensioned sketches of the following parts:

(1) Mild-steel crosshead; cottered bolt and cotter.

(2) Stud holding the U leather in position.

(3) C.I. cylinder gland and studs, allowing for 7 turns of $\frac{3}{4}''$ hemp packing.

(4) Holding down brackets, which are to be of $\frac{7}{8}''$ metal, and cast solid with the cylinder. There are 2 brackets, one at each end of cylinder, and the minimum shear load is to be assumed to be carried by one bracket only. The diameter of bolts required is to be found. (Actual strength calculations for the brackets are not required, but the method of checking the stress is to be indicated.) Approximate stresses:

Bearing pressure on side rod bushes: 2500 lb./in.$^2$.

Tensile stress for cast iron: 3000 lb./in.$^2$.

Tensile stress for mild steel: 10,000 lb./in.$^2$.

(Use proportional values for shear and crushing stresses.)

Load table for studs:

| | | | | | | |
|---|---|---|---|---|---|---|
| Dia., in. | $\frac{7}{8}$ | 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $1\frac{3}{8}$ | $1\frac{1}{2}$ |
| Load, lb. | 2900 | 4200 | 5700 | 7700 | 9400 | 11,600 |

*Glas. Univ., B.Sc.*

**(2)** Some particulars of a **spring-loaded valve** are given in fig. 2. Design the spring and complete the table given. Also design the cast-iron bridge. The valve is to open at 39 lb./in.$^2$ pressure. Diameter of valve seat = $7\frac{1}{2}''$. Valve lift = $1\frac{3}{4}''$. The compression in the spring, with valve closed, is $2\frac{13}{16}''$. Take tensile stress for C.I. = 2000 lb./in.$^2$. For strength and stiffness of spring take:

$$S = \sqrt[3]{\frac{W.D}{13{,}500}} \text{ and } d = \frac{W.D^3.N}{2 \times 10^6 \times S^4},$$

where W = load (lb.),

D = mean dia. of coils (inches),

S = side of square section (inches),

N = number of free coils (total number minus one),

$d$ = compression (inches) for load W.

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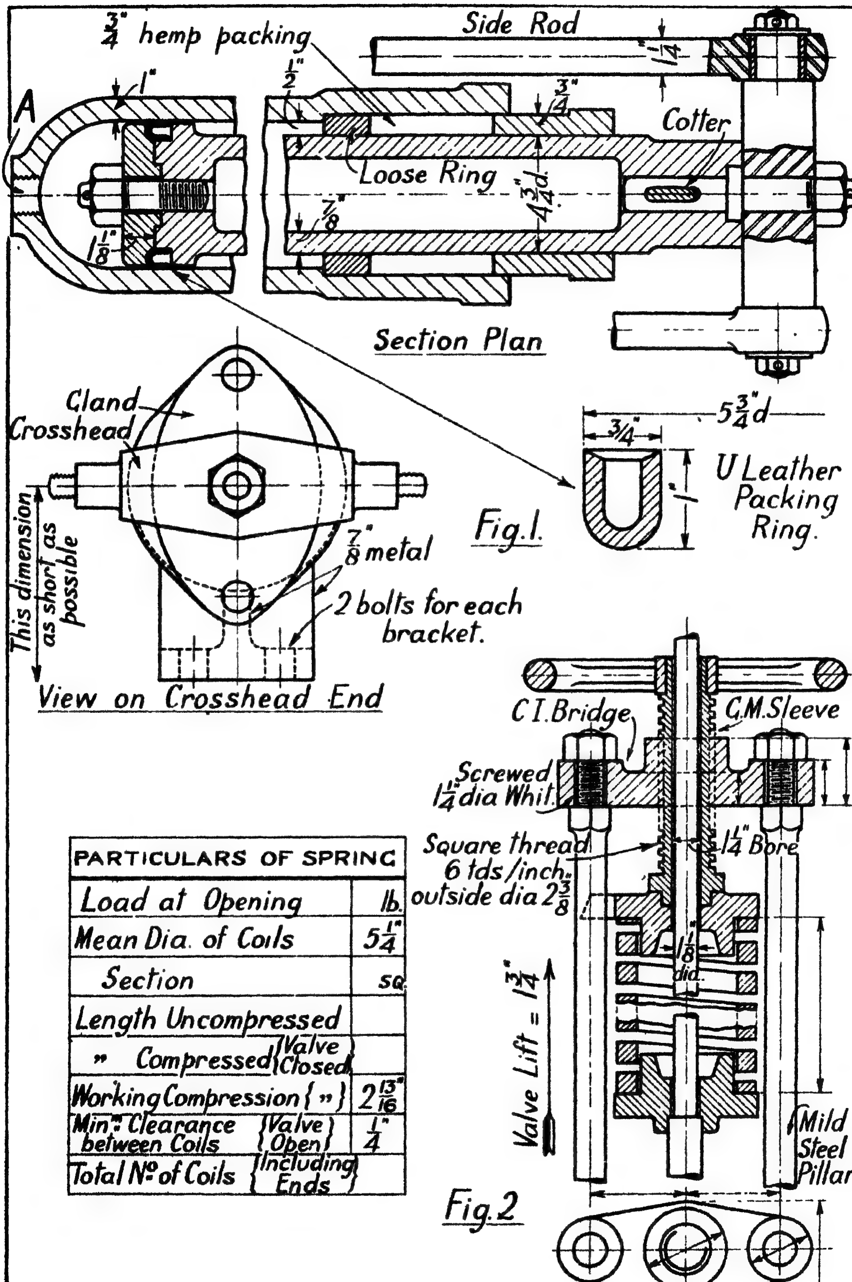
**Note.**—For the above hand sketches only were required, approximately to scale, and properly dimensioned. Time for question (1) about 70 min.; for question (2) about 35 min.




| PARTICULARS OF SPRING | |
|---|---|
| Load at Opening | lb. |
| Mean Dia. of Coils | 5 1/4" |
| Section | sq. |
| Length Uncompressed | |
| " Compressed {Valve Closed} | |
| Working Compression { " } | 2 13/16" |
| Min.m Clearance between Coils {Valve Open} | 1/4" |
| Total No. of Coils {Including Ends} | |

(1) Design the large end of a **connecting rod** of the type shown in the sketch, fig. 1, for a vertical engine, to the following particulars: Maximum load on rod: 52,000 lb. Diameter of connecting rod near large end: $5\frac{1}{4}''$. Diameter of crank pin: $11\frac{1}{2}''$. Tensile stress in bolts not more than 6000 lb./in.$^2$. Adopt any other values required, and give all necessary calculations. Make a working drawing of the rod end, half full size.

*Lond. Univ., B.Sc.*

(2) Design, and draw full size, a **relief valve** for the I.P. cylinder of a large vertical engine. The sketch, fig. 2, shows diagrammatically the design adopted for the H.P. cylinder. The new design is to be similar in form, and to the following particulars. The horizontal part of the valve body is to be $2''$ bore, and the vertical bore at the valve seat $2\frac{1}{2}''$ dia. The L-shaped casting is to have a body thickness of $\frac{1}{2}''$, and is to carry the provisions for the cylinder drain and indicator, screwed $1\frac{3}{8}''$ and $1\frac{1}{8}''$ respectively. The load on the spring is to be applied to the top of the valve by a vertical rod, $\frac{3}{4}''$ dia., the upper end of which is guided by the hollow adjusting screw, which presses upon the upper one of the two loose collars for holding the spring in place. The clearance, E, above the vertical rod is to be sufficient to allow for the full lift of the valve when the adjusting screw is screwed down into contact with its lock nut.

The square steel spring is to be designed with a mean diameter, D, of the coils $2\frac{1}{2}''$, and is to hold the valve to its seat against any pressure not exceeding 75 lb./in.$^2$ gauge pressure on the underside of the valve. The number of free coils of the spring is to be sufficient to allow the valve to lift not less than $\frac{1}{8}''$ from its seat when the pressure on the underside of the valve is increased to 85 lb./in.$^2$ gauge pressure. The working load on the spring, when the valve is closed, may be taken as: $11{,}000\ S^3 \div D$; and the deflection as $5{\cdot}584\ W.N.D^3 \div G.S^4$; where D = the mean diameter of the coils, N = number of free coils, W = the load on the spring, S = side of square steel, and the Modulus of Rigidity, G, may be taken as 12,000,000 lb./in.$^2$.

*Lond. Univ., B.Sc.*

(3) A quadruple riveted double **butt strap joint** with scalloped cover plates, as shown in fig. 3, is to be used for the longitudinal joint of a steam boiler drum 5 ft. dia., the working pressure being 450 lb./in.$^2$. The efficiency of the joint is to be 94 per cent. Calculate the plate thickness for steel of 28 ton/in.$^2$ ultimate strength, and a factor of safety not less than 4. Then design and make a completely dimensioned drawing of the joint, with the aid of the proportional pitches given. Fix the rivet diameter so that the plate and rivet efficiencies may be nearly equal. Assume the ratio of tensile to shear stress as 28/23, and use the standard expression for cover plate thickness

$$t_1 = \frac{5}{8}\left(\frac{p-d}{p-nd}\right)t,$$

where $n$ is the number of inner row rivets per wide pitch $p$. Assume that $1{\cdot}875 \times$ rivet area is available in double shear.

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**Note.**—Time for questions (1) and (2), 3 hr. each; for question (3), about 35 min.

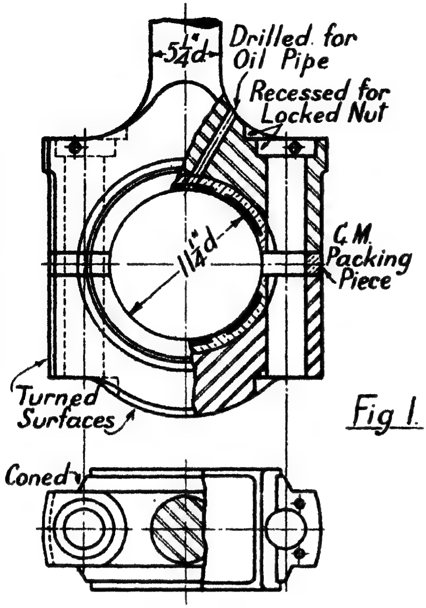


Fig 1.

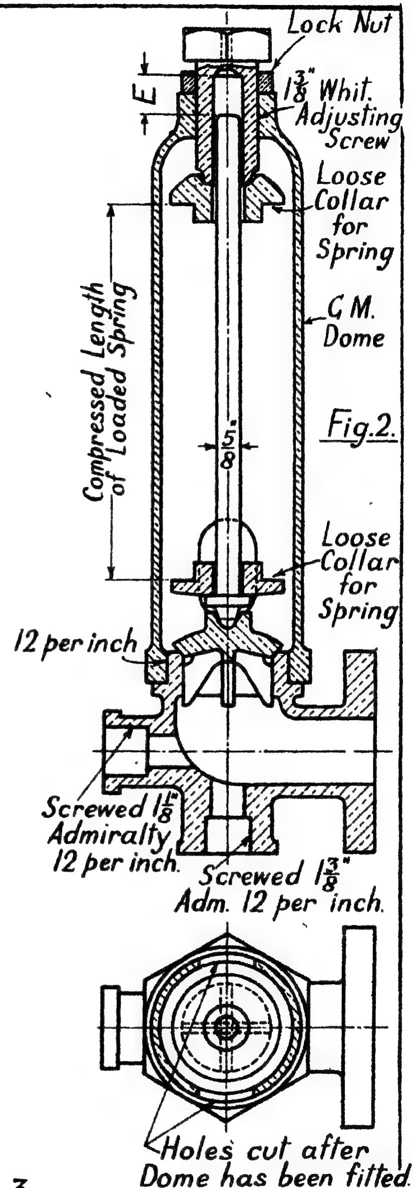


Fig.2.

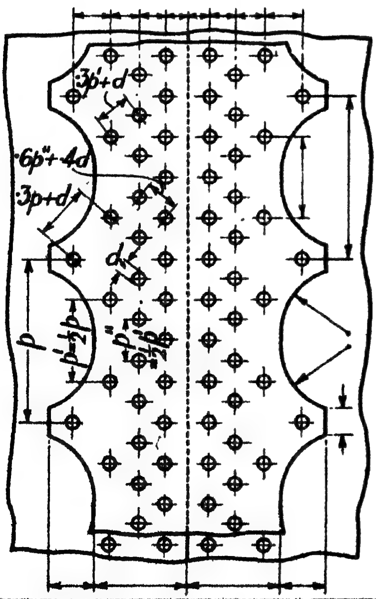


Fig.3.

**(1)** A small **hydraulic cylinder and ram** with a two-to-one multiplying gear are shown in fig. 1. A pull of 6500 lb. is required on the free end of the rope when the working pressure is 1200 lb./in.$^2$. The efficiency of the arrangement may be assumed as 90 per cent.

Make the calculations specified below and draw, half size, (*a*) that part of the sectional elevation to the left of XX, (*b*) the view looking on the cover, (*c*) the view looking on the gland. Calculations for the following dimensions are required: Diameter of ram, thickness of cylinder, diameter of the studs holding on the cylinder cover and the gland, thickness of cylinder cover and gland flange, diameter and length of pulley spindle. The studs for the gland and the cylinder cover should be chosen from the following table, in which allowance has been made for "tightening up" stresses:

| | | | | | | |
|---|---|---|---|---|---|---|
| Dia. of stud, in. | $\frac{3}{4}$ | $\frac{7}{8}$ | 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $1\frac{3}{8}$ |
| Safe load in lb. | 1790 | 2900 | 4220 | 5700 | 7700 | 9400 |

The following may be taken as safe working stresses:

Tensile stress for cylinder barrel, 4000 lb./in.$^2$.

Tensile stress for cover and gland flange, 3000 lb./in.$^2$.

Tensile stress for pulley spindle, 9000 lb./in.$^2$.

Bearing pressure on pulley spindle, 1500 lb./in.$^2$.

*Lond. Univ., B.Sc.*

**(2)** *No figure.*—Design a **Hooke's Joint** or Universal Coupling to connect two shafts running at 120 revs./min. and transmitting 15 h.p. Fully dimensioned drawings are required. State clearly your assumptions.

*Lond. Univ., B.Sc.*

**(3)** *No figure.*—Design and draw the **circumferential and longitudinal joints** in the shell of a steam drum of 5 ft. 6 in. internal diameter. The steam pressure is 150 lb./in.$^2$ by gauge.

Choose your working stresses, and state them clearly on the drawing.

The longitudinal joint is to be of the double-cover plate type.

*Lond. Univ., B.Sc.*

**(4)** *No figure.*—Design the **big end of a connecting rod** which is to be capable of withstanding a maximum load of 40,000 lb. There must be a satisfactory arrangement for adjusting the brasses. The crank pin is overhung. Give a sectional elevational and plan.

*Lond. Univ., B.Sc.*

**(5)** *No figure.*—Design a **ring-oiled pedestal bearing** for a 2" shaft, running at 500 revs./min. The total load is 1000 lb. and the height of the centre of the shaft is 6". Fully dimensioned drawings are required, along with all relevant calculations.

*Lond. Univ., B.Sc.*

**(6)** *No figure.*—Design and draw a **spur wheel** of 24" diameter capable of transmitting 50 h.p. at 120 revs./min. The shaft is 4" diameter and the teeth are of cast steel, machine cut, and of involute section. Include a full scale drawing of the tooth profile.

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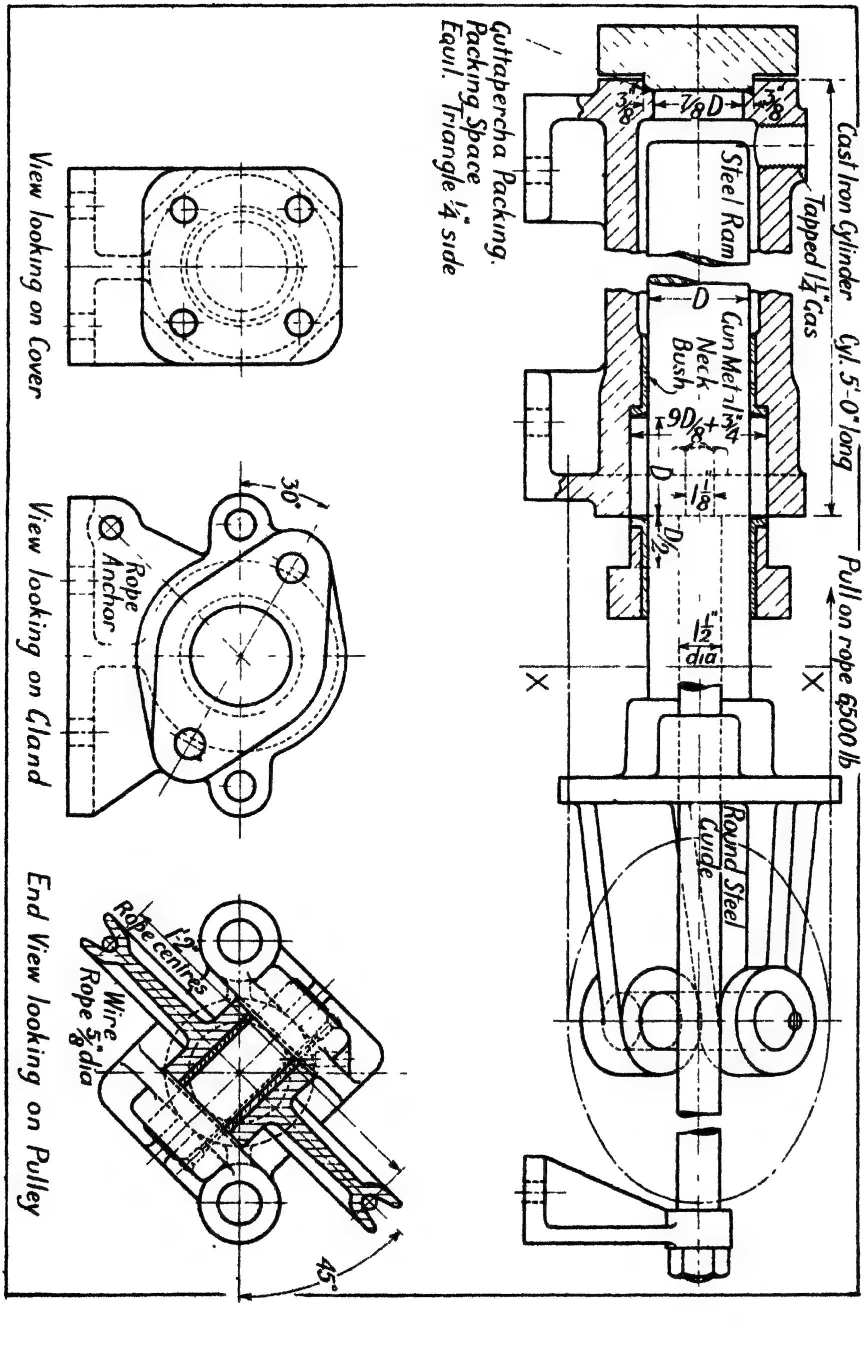
**Note.**—Time allowed for each of the above, 3 hr.


Cast Iron Cylinder
Tapped 1¼" Gas
Cyl. 5'-0" long
Pull on rope 6500 lb
Steel Ram
Gun Metal Neck Bush
Round Steel Guide
Guttapercha Packing.
Packing Space
Equil. Triangle ¼" side
View looking on Cover
View looking on Gland
Rope Anchor
End View looking on Pulley
Rope centres
Wire Rope ⅝" dia
30°
45°

# APPENDIX

## MATERIALS *

**Aluminium, Al; Antimony, Sb; Carbon, C; Chromium, Cr; Copper, Cu; Iron, Fe; Lead, Pb; Magnesium, Mg; Manganese, Mn; Nickel, Ni; Phosphorus, P; Silicon, Si; Tin, Sn; Zinc, Zn.**

| Name. | Process, Composition, &c. | Properties and Uses. |
|---|---|---|
| **Cast Iron (C.I.)** | Produced from calcined iron ore by the blast furnace. Varies from | |
| | **Grey C.I.** Combined C 0·1; Graphitic C 3·4; Si 2·5 | Soft, easily machined: for general casting purposes—Cylinders, Machine Tool Beds, &c. |
| | to **White C.I.** Combined C 3·2; Graphitic C 0·1; Si—less than 0·8 | Hard and brittle: for conversion into wrought iron. |
| | **Malleable C.I.** Formed by heating white C.I. for prolonged periods in association with iron oxides. | Tougher and more malleable than ordinary castings. Suitable for Levers, Pipe Fittings, &c. |
| | **Perlit.** Grey C.I. with low Si content is cast in moulds preheated up to as much as 500° C. | Special growth-resisting iron for high temperatures. Suitable for DieselEngineCylinders,Pistons,&c. |
| | **Semi-steel.** Obtained by charging scrap steel with the pig iron in the cupola before casting. | Tougher and more malleable than ordinary castings. |
| **Wrought Iron (W.I.)** | Obtained by eliminating most of the C and other impurities from white C.I., principally by the Puddling Process. W.I. is nearly pure and hence is not easily fused. | Moderately hard, strong, and tough. Can be welded. Used for Chains, Furnace Bars and Plates, Bolts, Rivets, &c. |
| **Steel.** | Contains from ·05 to 2·2 per cent C and is produced from C.I. by the Bessemer and Open-hearth Processes, and from W.I. by the Cementation and Crucible Processes. | Properties depend mainly on the C content. Increase of C gives increased hardness but decreased ductility. |
| | **Mild Steel** Under ·1 per cent C (dead mild). | Wire Rod. |
| | ·1 to ·3 per cent C (mild). | Boiler Plates, Structural Steel. |
| | ·3 to ·5 per cent C (medium). | Crank shafts, Connecting Rods, Springs. |
| | **Hard Steel.** Over ·5 per cent C. | Cutting Tools, Dies. |
| **Alloy Steels.** | **Nickel Steel.** 2 to 3·75 per cent Ni. | Toughness and shock-resisting properties. Used for Propeller Shafts, Axles, Gearing. |
| | **Nickel Steel (Invar).** 36 per cent Ni. | Non-expansive. |
| | **Nickel Chrome Steel.** | Toughness and hardness. Used for Turbine Discs. |
| | **Stainless Steel.** ·2 to ·4 per cent C, 12 to 14 per cent Cr., over ·2 per cent Si. | For Turbine Blades. |
| **Copper.** | Used chiefly alloyed with other metals. | |
| **Copper Alloys.** | **Brass.** Best quality, 70 per cent Cu, 30 per cent Zn. | |
| | **Muntz Metal.** 60 per cent Cu, 40 per cent Zn. | Can be rolled and forged when hot. |

## MATERIALS (*Continued*)

| Name. | Process, Composition, &c. | Properties and Uses. |
|---|---|---|
| **Copper Alloys.** | **Naval Brass:** 62% Cu, 37% Zn, 1% Sn. 70% Cu, 29% Zn, 1% Sn. | Can be rolled and forged when hot. For Condenser and other Tubes. |
| | **Bronze or G.M.** 80 to 90 per cent Cu, rest Sn. **Admiralty G.M.** 88 per cent Cu, 10 per cent Sn, 2 per cent Zn. | For Castings generally. |
| | **Manganese Bronze.** Bronze with the addition of a little ferro-manganese. | Gives good Castings of great strength. Used for ships' Propellers, large Bearings, &c. |
| | **Manganese Copper.** 3 to 4 per cent Mn, rest Cu. | For Turbine Blades. |
| | **Phosphor Bronze.** 90 to 96 per cent Cu, 10 to 4 per cent Sn, and above 0·1 per cent P. | Hardness and toughness. Gives good Castings. |
| | **Aluminium Bronze.** 90 to 95 per cent Cu, rest Al. | Non-corrodible in sea-water. Tenacity and toughness. |
| **Aluminium Alloys.** | **Aluminium Silicon.** 9 to 14 per cent Si, traces of Fe and Mn, remainder Al. | For light Castings generally. |
| | **Duralumin.** 4% Cu, 1% Mg, traces of Fe and Mn, remainder Al. | Tenacity equal to that of steel. Can be forged. |
| | **Y Alloy.** 4 per cent Cu, 2 per cent Ni, 1·5 per cent Mg, 92·5 per cent Al. | Retains strength at high temperatures. Used for Aero-engine Pistons, &c. |
| **White Metal.** | 87 per cent Sn, 7·8 per cent Cu, 5·2 per cent Sb. Other alloys contain less tin and more antimony, together with lead. | Babbitt's metal for Bearings. |

* Detailed specifications for many of the foregoing alloys have been published by the British Standards Institution.

## PROPERTIES OF MATERIALS

*The values given are average values and are in lb./in.²*

| | Ultimate Strength. | | | Limit of Proportionality. | Young's Modulus. |
|---|---|---|---|---|---|
| | Tension. | Compression. | Shear. | | |
| Cast Iron, Grey | 17,500 | 95,000 | 11,000 | 10,500 | $17 \times 10^6$ |
| Cast Iron, Malleable | 45,000 | — | — | — | — |
| Wrought Iron | 50,000 | 50,000 | 40,000 | 30,000 | $29 \times 10^6$ |
| Steel: | | | | | |
| Mild, ·1% C | 60,000 | 60,000 | 50,000 | 36,000 | |
| Medium, ·3 to ·5% C, annealed | 70,000 | 70,000 | 56,000 | 45,000 | |
| Medium, ·3 to ·5% C, tempered | 100,000 | — | — | 65,000 | |
| Hard, ·5 to ·7% C | 75,000 | 75,000 | 55,000 | 45,000 | $30 \times 10^6$ |
| Cast | 120,000 | — | — | 80,000 | |
| Steel Castings | 76,000 | — | — | 30,000 | |
| Nickel Steel | 80,000 | — | — | 60,000 | |
| Nickel Chrome Steel, quenched and tempered | 280,000 | — | — | 200,000 | |
| Brass, cast | 25,000 | — | — | — | $13·5 \times 10^6$ |
| Naval Brass | 50,000 | — | — | — | — |
| Gunmetal or Bronze | 30,000 | — | — | 6,000 | $13·5 \times 10^6$ |
| Manganese Bronze | 65,000 | — | — | — | — |
| Phosphor Bronze (rolled) | 60,000 | 60,000 | 40,000 | 19,000 | $14 \times 10^6$ |
| Phosphor Bronze (cast) | 35,000 | — | — | — | — |
| Aluminium, cast | 13,000 | — | — | — | — |
| Duralumin, heat treated | 50,000 | — | — | — | $11 \times 10^6$ |

A Table of Factors of Safety is given on p. 222.

## FACTORS OF SAFETY

*Average values of the ratio U.T.S. ÷ working stress*

| Material. | Kind of Load. | | | |
|---|---|---|---|---|
| | Steady. | Varying. | | Shock. |
| | | Stress always of the same kind. | Stress Alternating. | |
| Cast Iron - - - | 4 | 6 | 10 | 15 |
| Wrought Iron - - | 4 | 6 | 8 | 12 |
| Mild Steel - - - | 4 | 6 | 8 | 12 |
| Cast Steel - - - | 5 | 6 | 8 | 15 |
| Brittle Alloys - - | 5 | 6 | 10 | 15 |
| Soft Alloys - - | 5 | 6 | 8 | 12 |
| Timber - - - | 6 | 10 | 14 | 20 |

## LIMITING BEARING PRESSURES

*The figures below give average values of limiting bearing pressures in pounds per square inch of projected area.*

Line Shaft Bearings: Grease Lubricated, 50.
Ring-oiled, G.M. or W.M., 200.

High-speed Bearings for Turbines, Motor Generators, &c., 100 to 150.*

Locomotive Axle Bearings, 300.

Thrust Bearings, Michell Type, 200.

Engine Bearings:

| | Main. | Crank. | Crosshead. | Guide. |
|---|---|---|---|---|
| Marine Engines - - | 500 | 650 | 1500 | 60 |
| Locomotive - - | 250 | 1500 | 3000 | 50 |
| Diesel - - - | 250 | 1500 | 2000 | 60 |

* In conjunction with a rubbing velocity of 130 to 100 ft./sec.

## STANDARD TABLES

The tables numbered 1 to 3 and 5 to 11 which follow have been extracted, by permission, from Reports published by the British Standards Institution. For further particulars reference should be made to the full Reports, the numbers of which are stated under each table. Official copies may be obtained from the Secretary, 28 Victoria Street, Westminster, S.W.1.

## Table 1.—BRITISH STANDARD WHITWORTH SCREW THREADS

| Full Diameter. | No. of Threads per Inch. | Core Diameter. | Cross Sectional Area at Bottom of Thread. | Full Diameter. | No. of Threads per Inch. | Core Diameter. | Cross Sectional Area at Bottom of Thread. |
|---|---|---|---|---|---|---|---|
| Inches. | (N.) | Inches. | Sq. In. | Inches. | (N.) | Inches. | Sq. In. |
| 1/4 | 20 | 0·1860 | 0·0272 | 1 3/4 | 5 | 1·4938 | 1·7528 |
| 5/16 | 18 | 0·2413 | 0·0457 | 2 | 4·5 | 1·7154 | 2·3111 |
| 3/8 | 16 | 0·2950 | 0·0683 | 2 1/4 | 4 | 1·9298 | 2·9249 |
| 7/16 | 14 | 0·3461 | 0·0941 | 2 1/2 | 4 | 2·1798 | 3·7318 |
| 1/2 | 12 | 0·3932 | 0·1214 | 2 3/4 | 3·5 | 2·3841 | 4·4641 |
| 9/16 | 12 | 0·4557 | 0·1631 | 3 | 3·5 | 2·6341 | 5·4496 |
| 5/8 | 11 | 0·5086 | 0·2032 | 3 1/4 | 3·25 | 2·8560 | 6·4063 |
| 11/16 | 11 | 0·5711 | 0·2562 | 3 1/2 | 3·25 | 3·1060 | 7·5769 |
| 3/4 | 10 | 0·6220 | 0·3039 | 3 3/4 | 3 | 3·3231 | 8·6732 |
| 7/8 | 9 | 0·7328 | 0·4218 | 4 | 3 | 3·5731 | 10·0272 |
| 1 | 8 | 0·8400 | 0·5542 | 4 1/2 | 2·875 | 4·0546 | 12·9118 |
| 1 1/8 | 7 | 0·9420 | 0·6969 | 5 | 2·75 | 4·5343 | 16·1477 |
| 1 1/4 | 7 | 1·0670 | 0·8942 | 5 1/2 | 2·625 | 5·0122 | 19·7301 |
| 1 1/2 | [illegible] | 1·2866 | 1·3001 | 6 | 2·5 | 5·4878 | 23·6521 |

Note.—Pitch in inches = 1 ÷ N.

[B.S.I. Publication No. 84.]

## Table 2.—BRITISH STANDARD FINE SCREW THREADS

| Full Diameter. | No. of Threads per Inch. | Core Diameter. | Cross Sectional Area at Bottom of Thread. | Full Diameter. | No. of Threads per Inch. | Core Diameter. | Cross Sectional Area at Bottom of Thread. |
|---|---|---|---|---|---|---|---|
| Inches. | (N.) | Inches. | Sq. In. | Inches. | (N.) | Inches. | Sq. In. |
| 3/16 | 32 | 0·1475 | 0·0171 | 1 1/4 | 9 | 1·1078 | 0·9639 |
| 7/32 | 28 | 0·1730 | 0·0235 | 1 3/8 | 8 | 1·2150 | 1·1593 |
| 1/4 | 26 | 0·2008 | 0·0317 | 1 1/2 | 8 | 1·3400 | 1·4100 |
| 9/32 | 26 | 0·2320 | 0·0423 | 1 5/8 | 8 | 1·4650 | 1·6860 |
| 5/16 | 22 | 0·2543 | 0·0508 | 1 3/4 | 7 | 1·5670 | 1·9285 |
| 3/8 | 20 | 0·3110 | 0·0760 | 2 | 7 | 1·8170 | 2·5930 |
| 7/16 | 18 | 0·3663 | 0·1054 | 2 1/4 | 6 | 2·0366 | 3·2580 |
| 1/2 | 16 | 0·4200 | 0·1385 | 2 1/2 | 6 | 2·2866 | 4·1065 |
| 9/16 | 16 | 0·4825 | 0·1828 | 2 3/4 | 6 | 2·5366 | 5·0540 |
| 5/8 | 14 | 0·5336 | 0·2236 | 3 | 5 | 2·7438 | 5·9133 |
| 11/16 | 14 | 0·5961 | 0·2791 | 3 1/4 | 5 | 2·9938 | 7·0399 |
| 3/4 | 12 | 0·6432 | 0·3249 | 3 1/2 | 4·5 | 3·2154 | 8·1201 |
| 13/16 | 12 | 0·7057 | 0·3911 | 3 3/4 | 4·5 | 3·4654 | 9·4319 |
| 7/8 | 11 | 0·7586 | 0·4520 | 4 | 4·5 | 3·7154 | 10·8418 |
| 1 | 10 | 0·8720 | 0·5972 | 4 1/4 | 4 | 3·9298 | 12·13 |
| 1 1/8 | 9 | 0·9828 | 0·7586 | | | | |

[B.S.I. Publication No. 84.]

## Table 3.—BRITISH ASSOCIATION SCREW THREADS

| Designating Number. | Full Diameter. | Pitch. | Core Diameter. | Cross Sectional Area at Bottom of Thread.* | Designating Number. | Full Diameter. | Pitch. | Core Diameter. | Cross Sectional Area at Bottom of Thread.* |
|---|---|---|---|---|---|---|---|---|---|
| | Mm. | Mm. | Mm. | Sq. Mm. | | Mm. | Mm. | Mm. | Sq. Mm. |
| 0 | 6·0 | 1·0 | 4·8 | 18·10 | 13 | 1·2 | 0·25 | 0·9 | 0·64 |
| 1 | 5·3 | 0·9 | 4·22 | 13·99 | 14 | 1·0 | 0·23 | 0·72 | 0·41 |
| 2 | 4·7 | 0·81 | 3·73 | 10·93 | 15 | 0·9 | 0·21 | 0·65 | 0·33 |
| 3 | 4·1 | 0·73 | 3·22 | 8·14 | 16 | 0·79 | 0·19 | 0·56 | 0·25 |
| 4 | 3·6 | 0·66 | 2·81 | 6·20 | 17 | 0·70 | 0·17 | 0·50 | 0·20 |
| 5 | 3·2 | 0·59 | 2·49 | 4·87 | 18 | 0·62 | 0·15 | 0·44 | 0·15 |
| 6 | 2·8 | 0·53 | 2·16 | 3·66 | 19 | 0·54 | 0·14 | 0·37 | 0·11 |
| 7 | 2·5 | 0·48 | 1·92 | 2·89 | 20 | 0·48 | 0·12 | 0·34 | 0·091 |
| 8 | 2·2 | 0·43 | 1·68 | 2·22 | 21 | 0·42 | 0·11 | 0·29 | 0·066 |
| 9 | 1·9 | 0·39 | 1·43 | 1·61 | 22 | 0·37 | 0·10 | 0·25 | 0·049 |
| 10 | 1·7 | 0·35 | 1·28 | 1·29 | 23 | 0·33 | 0·09 | 0·22 | 0·038 |
| 11 | 1·5 | 0·31 | 1·13 | 1·00 | 24 | 0·29 | 0·08 | 0·19 | 0·028 |
| 12 | 1·3 | 0·28 | 0·96 | 0·72 | 25 | 0·25 | 0·07 | 0·17 | 0·023 |

* These areas are approximate only.

Note.—1 mm. = 0·039″ approx.

[B.S.I. Publication No. 93.]

## Table 4.—AMERICAN NATIONAL COARSE THREADS

| No. or Full Diameter. | No. of Threads per Inch. | Core Diameter. | Cross Sectional Area at Bottom of Thread. | Full Diameter. | No. of Threads per Inch. | Core Diameter. | Cross Sectional Area at Bottom of Thread. |
|---|---|---|---|---|---|---|---|
| Inches. | | Inches. | Sq. In. | Inches. | | Inches. | Sq. In. |
| 1 | 64 | 0·0527 | 0·0022 | $\frac{7}{8}$ | 9 | 0·731 | 0·419 |
| 2 | 56 | 0·0628 | 0·0031 | 1 | 8 | 0·838 | 0·551 |
| 3 | 48 | 0·0719 | 0·0041 | $1\frac{1}{8}$ | 7 | 0·939 | 0·693 |
| 4 | 40 | 0·0795 | 0·0050 | $1\frac{1}{4}$ | 7 | 1·064 | 0·890 |
| 5 | 40 | 0·0925 | 0·0067 | $1\frac{3}{8}$ | 6 | 1·158 | 1·054 |
| 6 | 32 | 0·0974 | 0·0075 | $1\frac{1}{2}$ | 6 | 1·283 | 1·294 |
| 8 | 32 | 0·1234 | 0·0120 | $1\frac{3}{4}$ | 5 | 1·490 | 1·744 |
| 10 | 24 | 0·1359 | 0·0145 | 2 | $4\frac{1}{2}$ | 1·711 | 2·300 |
| 12 | 24 | 0·1619 | 0·0206 | $2\frac{1}{4}$ | $4\frac{1}{2}$ | 1·961 | 3·021 |
| $\frac{1}{4}$ | 20 | 0·185 | 0·027 | $2\frac{1}{2}$ | 4 | 2·175 | 3·716 |
| $\frac{5}{16}$ | 18 | 0·240 | 0·045 | $2\frac{3}{4}$ | 4 | 2·425 | 4·619 |
| $\frac{3}{8}$ | 16 | 0·294 | 0·068 | 3 | 4 | 2·675 | 5·621 |
| $\frac{7}{16}$ | 14 | 0·345 | 0·093 | $3\frac{1}{4}$ | 4 | 2·925 | 6·720 |
| $\frac{1}{2}$ | 13 | 0·400 | 0·126 | $3\frac{1}{2}$ | 4 | 3·175 | 7·918 |
| $\frac{9}{16}$ | 12 | 0·454 | 0·162 | $3\frac{3}{4}$ | 4 | 3·425 | 9·214 |
| $\frac{5}{8}$ | 11 | 0·507 | 0·202 | 4 | 4 | 3·675 | 10·608 |
| $\frac{3}{4}$ | 10 | 0·620 | 0·302 | | | | |

## Table 5.—BRITISH STANDARD PIPE THREADS (TAPER)

Refer to Fig. 1, Page 59

| 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|
| Nominal Bore of Tube. | Approx. Outside Diameter of Black Tube. | Gauge Diameter. | Number of Threads per Inch. | Length of Thread. On Pipe-end, Min. | Length of Thread. In Coupler, Min. | Distance of Gauge Diameter from Pipe-end (Taper Screw). |
| Inches. | Inches. | Inches. | | Inches. | Inches. | Inches. |
| 1/8 | 13/32 | 0·383 | 28 | 3/8 | 5/8 | 5/32 |
| 1/4 | 17/32 | 0·518 | 19 | 7/16 | 7/8 | 3/16 |
| 3/8 | 11/16 | 0·656 | 19 | 1/2 | 1 | 1/4 |
| 1/2 | 27/32 | 0·825 | 14 | 5/8 | 1 1/4 | 1/4 |
| 5/8 | 29/32 | 0·902 | 14 | 5/8 | 1 1/4 | 1/4 |
| 3/4 | 1 1/16 | 1·041 | 14 | 3/4 | 1 3/8 | 3/8 |
| 7/8 | 1 7/32 | 1·189 | 14 | 3/4 | 1 3/8 | 3/8 |
| 1 | 1 11/32 | 1·309 | 11 | 7/8 | 1 5/8 | 3/8 |
| 1 1/4 | 1 11/16 | 1·650 | 11 | 1 | 2 | 1/2 |
| 1 1/2 | 1 29/32 | 1·882 | 11 | 1 | 2 | 1/2 |
| 1 3/4 | 2 5/32 | 2·116 | 11 | 1 1/8 | 2 1/4 | 5/8 |
| 2 | 2 3/8 | 2·347 | 11 | 1 1/8 | 2 1/4 | 5/8 |
| 2 1/4 | 2 5/8 | 2·587 | 11 | 1 1/4 | 2 1/2 | 11/16 |
| 2 1/2 | 3 | 2·960 | 11 | 1 1/4 | 2 1/2 | 11/16 |
| 2 3/4 | 3 1/4 | 3·210 | 11 | 1 3/8 | 2 3/4 | 13/16 |
| 3 | 3 1/2 | 3·460 | 11 | 1 3/8 | 2 3/4 | 13/16 |
| 3 1/4 | 3 3/4 | 3·700 | 11 | 1 1/2 | 3 | 7/8 |
| 3 1/2 | 4 | 3·950 | 11 | 1 1/2 | 3 | 7/8 |
| 3 3/4 | 4 1/4 | 4·200 | 11 | 1 1/2 | 3 | 7/8 |
| 4 | 4 1/2 | 4·450 | 11 | 1 5/8 | 3 1/4 | 1 |
| 4 1/2 | 5 | 4·950 | 11 | 1 5/8 | 3 1/4 | 1 |
| 5 | 5 1/2 | 5·450 | 11 | 1 3/4 | 3 1/2 | 1 1/8 |
| 5 1/2 | 6 | 5·950 | 11 | 1 7/8 | 3 3/4 | 1 1/4 |
| 6 | 6 1/2 | 6·450 | 11 | 2 | 4 | 1 3/8 |
| 7 | 7 1/2 | 7·450 | 10 | 2 1/8 | 4 1/4 | 1 3/8 |
| 8 | 8 1/2 | 8·450 | 10 | 2 1/4 | 4 1/2 | 1 1/2 |
| 9 | 9 1/2 | 9·450 | 10 | 2 1/4 | 4 1/2 | 1 1/2 |

[B.S.I. Publication No. 21.]

## Table 6.—BRITISH STANDARD BRIGHT HEXAGON BOLT-HEADS AND NUTS

All Dimensions are in Inches. The use of Col. A gives the original Whitworth sizes. The use of Col. B gives the War Emergency British Standard.

| Diameter of Bolt. A | Diameter of Bolt. B | Hexagon Dimensions (max.). Width across Flats. | Hexagon Dimensions (max.). App. Width across Corners. | Bolt-head Thickness (max.) |
|---|---|---|---|---|
| — | 1/4 | 0·445 | 0·51 | 0·19 |
| 1/4 | 5/16 | 0·525 | 0·61 | 0·22 |
| 5/16 | 3/8 | 0·600 | 0·69 | 0·27 |
| 3/8 | 7/16 | 0·710 | 0·82 | 0·33 |
| 7/16 | 1/2 | 0·820 | 0·95 | 0·38 |
| 1/2 | 9/16 | 0·920 | 1·06 | 0·44 |
| 9/16 | 5/8 | 1·010 | 1·17 | 0·49 |
| 5/8 | 11/16 | 1·100 | 1·27 | 0·55 |
| 11/16 | 3/4 | 1·200 | 1·39 | 0·60 |
| 3/4 | 7/8 | 1·300 | 1·50 | 0·66 |
| 7/8 | 1 | 1·480 | 1·71 | 0·77 |
| 1 | 1 1/8 | 1·670 | 1·93 | 0·88 |
| 1 1/8 | 1 1/4 | 1·860 | 2·15 | 0·98 |
| 1 1/4 | 1 3/8 | 2·050 | 2·37 | 1·09 |
| 1 3/8 | 1 1/2 | 2·220 | 2·56 | 1·20 |
| 1 1/2 | 1 5/8 | 2·410 | 2·78 | 1·31 |
| 1 5/8 | 1 3/4 | 2·580 | 2·98 | 1·42 |
| 1 3/4 | 2 | 2·760 | 3·19 | 1·53 |
| 2 | — | 3·150 | 3·64 | 1·75 |

Nut Thickness: For A (Whitworth) = Dia. of Bolt; for B = 7/8 × Dia. of Bolt.

[B.S.I. Publications Nos. 190 and 1083.]

TABLE 7.—BRITISH STANDARD PIPE FLANGES (FOR LAND USE)

For Working Steam Pressure ($a$) up to 50 lb./in.$^2$; ($b$) above 50 and up to 100 lb./in.$^2$

(For working gas pressures up to 30 lb./in.$^2$ and for working water pressures up to 175 lb./in.$^2$ refer to B.S. No. 10, Part 1)

| Nominal Pipe Size. | Actual Outside Diameter of Wrought Pipe. | Diameter of Flange. | Diameter of Bolt Circle. | Number of Bolts. | | Diameter of Bolts. | | Thickness of Flange. Cast Iron. | | Thickness of Flange. Cast Steel and Bronze. | | Thickness of Flange. Iron or Steel (Stamped or Forged), Screwed or Riveted on with Boss or Welded on with Fillet. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | 50 lb. | 100 lb. | 50 lb. | 100 lb. | 50 lb. | 100 lb. | 50 lb. | 100 lb. | 50 lb. | 100 lb. |
| In. | In. | In. | In. | | | In. | In. | In. | In. | In. | In. | In. | In. |
| 1/2 | 27/32 | 3 3/4 | 2 5/8 | 4 | 4 | 1/2 | 1/2 | 1/2 | 1/2 | 3/8 | 3/8 | 3/16 | 1/4 |
| 3/4 | 1 1/16 | 4 | 2 7/8 | 4 | 4 | 1/2 | 1/2 | 1/2 | 1/2 | 3/8 | 3/8 | 3/16 | 1/4 |
| 1 | 1 11/32 | 4 1/2 | 3 1/4 | 4 | 4 | 1/2 | 1/2 | 1/2 | 1/2 | 3/8 | 3/8 | 3/16 | 9/32 |
| 1 1/4 | 1 21/32 | 4 3/4 | 3 7/16 | 4 | 4 | 1/2 | 1/2 | 5/8 | 5/8 | 1/2 | 1/2 | 1/4 | 5/16 |
| 1 1/2 | 1 29/32 | 5 1/4 | 3 7/8 | 4 | 4 | 1/2 | 1/2 | 5/8 | 5/8 | 1/2 | 1/2 | 1/4 | 11/32 |
| 2 | 2 3/8 | 6 | 4 1/2 | 4 | 4 | 5/8 | 5/8 | 3/4 | 3/4 | 9/16 | 9/16 | 5/16 | 3/8 |
| 2 1/2 | 3 | 6 1/2 | 5 | 4 | 4 | 5/8 | 5/8 | 3/4 | 3/4 | 9/16 | 9/16 | 5/16 | 13/32 |
| 3 | 3 1/2 | 7 1/4 | 5 3/4 | 4 | 4 | 5/8 | 5/8 | 3/4 | 3/4 | 9/16 | 9/16 | 3/8 | 7/16 |
| 3 1/2 | 4 | 8 | 6 1/2 | 4 | 8 | 5/8 | 5/8 | 3/4 | 3/4 | 9/16 | 9/16 | 3/8 | 15/32 |
| 4 | 4 1/2 | 8 1/2 | 7 | 4 | 8 | 5/8 | 5/8 | 7/8 | 7/8 | 11/16 | 11/16 | 3/8 | 1/2 |
| 5 | 5 1/2 | 10 | 8 1/4 | 8 | 8 | 5/8 | 3/4 | 7/8 | 7/8 | 11/16 | 11/16 | 1/2 | 9/16 |
| 6 | 6 1/2 | 11 | 9 1/4 | 8 | 8 | 5/8 | 3/4 | 7/8 | 7/8 | 11/16 | 11/16 | 1/2 | 11/16 |
| 7 | 7 1/2 | 12 | 10 1/4 | 8 | 8 | 5/8 | 3/4 | 1 | 1 | 3/4 | 3/4 | 1/2 | 3/4 |
| 8 | 8 1/2 | 13 1/4 | 11 1/2 | 8 | 8 | 5/8 | 3/4 | 1 | 1 | 3/4 | 3/4 | 1/2 | 3/4 |
| 9 | 9 1/2 | 14 1/4 | 12 3/4 | 8 | 12 | 5/8 | 3/4 | 1 | 1 | 3/4 | 13/16 | 5/8 | 13/16 |
| 10 | 10 1/2 | 16 | 14 | 8 | 12 | 3/4 | 3/4 | 1 | 1 | 3/4 | 7/8 | 5/8 | 7/8 |
| 12 | 12 1/2 | 18 | 16 | 12 | 12 | 3/4 | 7/8 | 1 1/8 | 1 1/8 | 7/8 | 1 | 5/8 | 1 |

Diameter of bolt holes: for 1/2″ and 5/8″ bolts, 1/16″ larger; for all other bolts, 1/8″ larger.

[B.S.I. Publication No. 10, Part 2.]

TABLE 8.—BRITISH STANDARD PIPE FLANGES (FOR LAND USE)

For Working Steam Pressures (*a*) above 100 and up to 150 lb./in.$^2$, (*b*) above 150 and up to 250 lb./in.$^2$, (*c*) above 250 and up to 350 lb./in.$^2$, (*d*) above 350 and up to 450 lb./in.$^2$ *

| Nominal Pipe Size. | Actual Outside Diameter of Wrought Pipe. | Diameter of Flange | | | Diameter of Bolt Circle. | | | Number of Bolts. | | Diameter of Bolts | | | | Thickness of Flange. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | | | Cast Iron. | Cast Steel and Bronze; Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet. | | | |
| | | 150 lb. | 250 lb. 350 lb. | 450 lb. | 150 lb. | 250 lb. 350 lb. | 450 lb. | 150 lb. 250 lb. 350 lb. | 450 lb. | 150 lb | 250 lb. | 350 lb. | 450 lb. | 150 lb. only. | 150 lb. | 250 lb. | 350 lb. | 450 lb. |
| In. | In. | In. | In. | In. | In. | In. | In. | | | In. | In. | In. | In. | In. | In. | In. | In. | In. |
| 1/2 | 27/32 | 3 3/4 | 4 1/2 | 4 1/2 | 2 5/8 | 3 1/4 | 3 1/4 | 4 | 4 | 1/2 | 5/8 | 5/8 | 5/8 | 1/2 | 3/8 | 1/2 | 5/8 | 3/4 |
| 3/4 | 1 1/16 | 4 | 4 1/2 | 4 1/2 | 2 7/8 | 3 1/4 | 3 1/4 | 4 | 4 | 1/2 | 5/8 | 5/8 | 5/8 | 1/2 | 3/8 | 1/2 | 5/8 | 3/4 |
| 1 | 1 11/32 | 4 1/2 | 4 3/4 | 5 | 3 7/16 | 3 7/16 | 3 3/4 | 4 | 4 | 5/8 | 5/8 | 5/8 | 5/8 | 1/2 | 3/8 | 9/16 | 3/4 | 7/8 |
| 1 1/4 | 1 21/32 | 5 1/4 | 5 1/4 | 5 1/4 | 3 7/8 | 3 7/8 | 3 7/8 | 4 | 4 | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 | 7/16 | 11/16 | 3/4 | 7/8 |
| 1 1/2 | 1 29/32 | 5 1/2 | 5 1/2 | 6 | 4 1/8 | 4 1/8 | 4 1/2 | 4 | 4 | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 | 1/2 | 11/16 | 7/8 | 1 |
| 2 | 2 3/8 | 6 1/2 | 6 1/2 | 6 1/2 | 5 | 5 | 5 | 4 | 8 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 5/8 | 3/4 | 1 | 1 |
| 2 1/2 | 3 | 7 1/2 | 7 1/2 | 7 1/2 | 5 3/4 | 5 3/4 | 5 3/4 | 8 | 8 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 5/8 | 3/4 | 1 | 1 1/8 |
| 3 | 3 1/2 | 8 | 8 | 8 | 6 1/4 | 6 1/4 | 6 1/2 | 8 | 8 | 5/8 | 5/8 | 3/4 | 3/4 | 3/4 | 5/8 | 7/8 | 1 1/4 | 1 1/4 |
| 3 1/2 | 4 | 8 1/2 | 8 1/2 | 9 | 7 | 7 | 7 1/2 | 8 | 8 | 5/8 | 5/8 | 3/4 | 7/8 | 7/8 | 3/4 | 7/8 | 1 1/4 | 1 1/4 |
| 4 | 4 1/2 | 9 | 9 | 9 1/2 | 7 1/4 | 7 1/4 | 7 3/4 | 8 | 8 | 5/8 | 5/8 | 3/4 | 7/8 | 7/8 | 3/4 | 1 | 1 3/8 | 1 3/8 |
| 5 | 5 1/2 | 11 | 11 | 11 | 9 1/4 | 9 1/4 | 9 1/4 | 8 | 12 | 3/4 | 3/4 | 7/8 | 7/8 | 1 | 7/8 | 1 1/8 | 1 3/8 | 1 5/8 |
| 6 | 6 1/2 | 12 | 12 | 12 | 10 1/4 | 10 1/4 | 10 1/4 | 12 | 12 | 3/4 | 3/4 | 7/8 | 7/8 | 1 | 7/8 | 1 1/8 | 1 3/8 | 1 5/8 |
| 7 | 7 1/2 | 13 1/4 | 13 1/4 | 13 1/4 | 11 1/2 | 11 1/2 | 11 1/2 | 12 | 12 | 3/4 | 3/4 | 7/8 | 1 | 1 | 7/8 | 1 1/4 | 1 5/8 | 1 3/4 |
| 8 | 8 1/2 | 14 1/2 | 14 1/2 | 14 1/2 | 12 3/4 | 12 3/4 | 12 1/2 | 12 | 12 | 3/4 | 3/4 | 7/8 | 1 | 1 1/8 | 1 | 1 1/4 | 1 5/8 | 1 7/8 |
| 9 | 9 1/2 | 16 | 16 | 16 | 14 | 14 | 14 | 12 | 16 | 7/8 | 7/8 | 1 | 1 | 1 1/8 | 1 | 1 3/8 | 1 3/4 | 2 |
| 10 | 10 1/2 | 17 | 17 | 17 | 15 | 15 | 15 | 12 | 16 | 7/8 | 7/8 | 1 | 1 | 1 1/8 | 1 | 1 3/8 | 1 7/8 | 2 |
| 12 | 12 1/2 | 19 1/4 | 19 1/4 | 19 1/4 | 17 1/4 | 17 1/4 | 17 | 16 | 16 | 7/8 | 7/8 | 1 | 1 1/8 | 1 1/4 | 1 1/8 | 1 1/2 | 2 | 2 1/4 |

Diameter of bolt holes: for 1/2" and 5/8" bolts, 1/16" larger; for all other bolts, 1/8" larger.

* For pressures from 450 to 600 lb./in.$^2$ refer to B.S.I. Publication No. 10, Part 3.

[B.S.I. Publication No. 10, Part 2.]

## Table 9.—BRITISH STANDARD HYDRAULIC POWER PIPES

All Dimensions are in Inches

The Letters refer to the Drawings on Page 61

| | | Class A. Working Pressure: 700–900 lb./sq. in. Test Pressure: 2500 lb./sq. in. Flange, Type 1. | | | Class A. Flange, Type 2. | | | | Class B. Working Pressure: 900–1200 lb./sq. in. Test Pressure: 3300 lb./sq. in. Flange, Type 1. | | Class B. Flange, Type 2. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Bore | A | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 2 | 3 | 4 | 5 | 6 | 7 |
| Flange length | B | 7½ | 9½ | 11½ | 14½ | 16 | 20 | 22 | 8¾ | 11¼ | 14 | 16½ | 19 | 21 |
| Spigot diameter | C | 2¾ | 3¼ | 4¼ | 5¾ | 7 | 8 | 9 | 2¾ | 3¾ | 4⅞ | 6 | 7 | 8 |
| Flange width | E | 3⅞ | 5¼ | 6½ | 7⅞ | 9 | 12 | 14 | 4¼ | 6 | 7½ | 9 | 10½ | 13 |
| Bolt centres | F | 5⅜ | 6¾ | 8¼ | 10½ | 11½ | 14½ | 16 | 6 | 8 | 10½ | 12½ | 14 | 15½ |
| Bolt holes | H | 1⅛ | 1¼ | 1⅜ | 1⅝ | 1¾ | 2⅛ | 2⅜ | 1¼ | 1⅜ | 1⅝ | 2 | 2¼ | 2½ |
| Spigot depth | J, K | ¼ | ¼ | ⅜ | ⅜ | ⅜ | ⅜ | ⅜ | ½ | ½ | ⅜ | ⅜ | ⅜ | ⅜ |
| Recess max. depth | $K_1$ | ⅜ | ⅜ | ½ | ½ | ½ | ½ | ½ | ⅜ | ⅜ | ½ | ½ | ½ | ½ |
| Flange set back | M | ¼ | ¼ | ¼ | 1¼ | 1¼ | 1⅛ | 1⅛ | ½ | ½ | 1 | 1¼ | 1⅜ | 1⅝ |
| Flange thickness | N | 1⅝ | 2¼ | 2½ | 2¾ | 3 | 3⅛ | 3¾ | 2¼ | 2¾ | 3 | 3½ | 3¾ | 3⅞ |
| Pipe thickness | T | 9/16 | ⅝ | 13/16 | 1 | 1⅛ | 1¼ | 1⅜ | ⅝ | ⅞ | 1 1/16 | 1¼ | 1½ | 1⅝ |
| Bolt diameter | | ⅞ | 1 | 1¼ | 1⅜ | 1⅝ | 2 | 2¼ | 1 | 1¼ | 1½ | 1¾ | 2 | 2¼ |
| Standard length of pipe | | 72 | 108 | 108 | 108 | 108 | 108 | 108 | 72 | 108 | 108 | 108 | 108 | 108 |

[B.S.I. Publication No. 44.]

Table 10.—*See opposite page.*

## Table 11.—BRITISH STANDARD WOODRUFF KEYS AND KEYWAYS

Refer to Fig. 7, Page 71. All Dimensions are in Inches

| B.S. Key Number. | Diameter of Key, *a*. | Depth of Key, *b*. | Thickness of Key, *c*. | Depth of Keyway in Shaft, *f*. | B.S. Key Number. | Diameter of Key, *a*. | Depth of Key, *b*. | Thickness of Key, *c*. | Depth of Keyway in Shaft, *f*. |
|---|---|---|---|---|---|---|---|---|---|
| 10 | ½ | 0·203 | 0·0635 | 0·1668 | 150 | 1 | 0·438 | 0·2510 | 0·3080 |
| 20 | ½ | 0·203 | 0·0948 | 0·1511 | 155 | 1 | 0·438 | 0·3135 | 0·2768 |
| 30 | ½ | 0·203 | 0·1260 | 0·1355 | 160 | 1⅛ | 0·484 | 0·1885 | 0·3853 |
| 40 | ⅝ | 0·250 | 0·0948 | 0·1981 | 180 | 1⅛ | 0·484 | 0·2510 | 0·3540 |
| 50 | ⅝ | 0·250 | 0·1260 | 0·1825 | 185 | 1⅛ | 0·484 | 0·3135 | 0·3228 |
| 60 | ⅝ | 0·250 | 0·1573 | 0·1669 | 210 | 1¼ | 0·547 | 0·2510 | 0·4170 |
| 70 | ¾ | 0·313 | 0·1260 | 0·2455 | 215 | 1¼ | 0·547 | 0·3135 | 0·3858 |
| 80 | ¾ | 0·313 | 0·1573 | 0·2299 | 225 | 1¼ | 0·547 | 0·3760 | 0·3545 |
| 90 | ¾ | 0·313 | 0·1885 | 0·2143 | 230 | 1⅜ | 0·594 | 0·3135 | 0·4328 |
| 100 | ⅞ | 0·375 | 0·1573 | 0·2919 | 235 | 1⅜ | 0·594 | 0·3760 | 0·4015 |
| 110 | ⅞ | 0·375 | 0·1885 | 0·2763 | 240 | 1½ | 0·641 | 0·2510 | 0·5110 |
| 115 | ⅞ | 0·375 | 0·2510 | 0·2450 | 250 | 1½ | 0·641 | 0·3135 | 0·4798 |
| 130 | 1 | 0·438 | 0·1885 | 0·3393 | 255 | 1½ | 0·641 | 0·3760 | 0·4485 |

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## TABLE 10.—BRITISH STANDARD RECTANGULAR PARALLEL KEYS

All Dimensions are in Inches

| Designation. | Shaft Diameters. Over. | Shaft Diameters. Up to and including. | Key. Nominal Width. | Key. Nominal Thickness. | Standard Lengths. Minimum. | Standard Lengths. Increasing by. | Standard Lengths. Maximum. |
|---|---|---|---|---|---|---|---|
| B.S.K. 3/32 R | 3/16 | 1/4 | 3/32 | 3/32 | 3/8 | 1/8 | 3/4 |
| „ 1/8 „ | 1/4 | 3/8 | 1/8 | 1/8 | 1/2 | 1/8 | 1 |
| „ 5/32 „ | 3/8 | 1/2 | 5/32 | 5/32 | 5/8 | 1/8 | 1 1/4 |
| „ 3/16 „ | 1/2 | 3/4 | 3/16 | 5/32 | 3/4 | 1/4 | 1 1/2 |
| „ 1/4 „ | 3/4 | 1 | 1/4 | 3/16 | 1 | 1/4 | 2 |
| „ 5/16 „ | 1 | 1 1/4 | 5/16 | 7/32 | 1 1/4 | 1/4 | 2 1/2 |
| „ 3/8 „ | 1 1/4 | 1 1/2 | 3/8 | 1/4 | 1 1/2 | 1/4 | 3 |
| „ 7/16 „ | 1 1/2 | 1 3/4 | 7/16 | 9/32 | 1 3/4 | 1/4 | 3 1/2 |
| „ 1/2 „ | 1 3/4 | 2 | 1/2 | 11/32 | 2 | 1/2 | 4 |
| „ 9/16 „ | 2 | 2 1/4 | 9/16 | 3/8 | 2 1/2 | 1/2 | 5 |
| „ 5/8 „ | 2 1/4 | 2 1/2 | 5/8 | 7/16 | 2 1/2 | 1/2 | 5 |
| „ 11/16 „ | 2 1/2 | 2 3/4 | 11/16 | 15/32 | 3 | 1/2 | 6 |
| „ 3/4 „ | 2 3/4 | 3 | 3/4 | 1/2 | 3 | 1/2 | 6 |
| „ 7/8 „ | 3 | 3 1/2 | 7/8 | 5/8 | 3 1/2 | 1/2 | 7 |
| „ 1 „ | 3 1/2 | 4 | 1 | 11/16 | 4 | 1 | 8 |
| „ 1 1/8 „ | 4 | 4 1/2 | 1 1/8 | 3/4 | 5 | 1 | 10 |
| „ 1 1/4 „ | 4 1/2 | 5 | 1 1/4 | 13/16 | 5 | 1 | 10 |
| „ 1 3/8 „ | 5 | 5 1/2 | 1 3/8 | 15/16 | 6 | 1 | 12 |
| „ 1 1/2 „ | 5 1/2 | 6 | 1 1/2 | 1 | 6 | 1 | 12 |
| „ 1 5/8 „ | 6 | 6 1/2 | 1 5/8 | 1 1/16 | 7 | 1 | 14 |
| „ 1 3/4 „ | 6 1/2 | 7 | 1 3/4 | 1 3/16 | 7 | 1 | 14 |
| „ 1 7/8 „ | 7 | 7 1/2 | 1 7/8 | 1 1/4 | 8 | 1 | 16 |
| „ 2 „ | 7 1/2 | 8 | 2 | 1 3/8 | 8 | 2 | 16 |
| „ 2 1/4 „ | 8 | 9 | 2 1/4 | 1 1/2 | 10 | 2 | 20 |
| „ 2 1/2 „ | 9 | 10 | 2 1/2 | 1 5/8 | 10 | 2 | 20 |
| „ 2 3/4 „ | 10 | 11 | 2 3/4 | 1 7/8 | 12 | 2 | 24 |
| „ 3 „ | 11 | 12 | 3 | 2 | 12 | 2 | 24 |

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